JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XXIII. FEBRUARY, 1911. No. 1.

The Society as a body is not responsible for statements made by individual members.

COUNCIL OF THE SOCIETY

(Under whose supervision this number is published).

Eng.-in-Chief H. I. CONE, U. S. N. Captain R. S. GRIFFIN, U. S. N.

Eng.-in-Chf. CHARLES A. MCALLISTER, U.S.R.C.S. Lieut. Commander U. T. HOLMES, U. S. N.

Lieut. Commander J. K. ROBISON, U. S. N.

DESCRIPTION AND TRIALS OF U. S. TORPEDO-BOAT DESTROYERS *ROE* AND *TERRY*.

BY C. F. BAILEY, ASSOCIATE.

The torpedo-boat destroyers No. 24, *Roe*, and No. 25, *Terry*, are two of the ten torpedo-boat destroyers, Nos. 22 to 31, bids for which were opened at the Navy Department, Washington, D. C., September 1, 1908. The contracts were awarded the Newport News Shipbuilding and Dry Dock Company, Newport News, Va., and were signed October 12, 1908. The vessels were to be completed and ready for delivery on or before October 12, 1910. The contract price for each of these vessels was $620,000, of which amount $360,000 was allotted for propelling machinery.

The keel of the *Roe* was laid January 18, 1909, and the vessel was launched July 24, 1909. The keel of the *Terry* was laid January 28, 1909, and the vessel was launched August 21, 1909. The preliminary official trials of the *Roe* were run July 12 to 16, 1910, and the trials of the *Terry* were run September 20 to 24, 1910. The *Roe* was completed and

delivered at the Norfolk Navy Yard, September 15, 1910, about one month ahead of the contract time. The *Terry* was delivered at the same yard October 12, 1910, the contract date of completion.

The general requirements of the contract were as follows:

Trial displacement, about, ...	742 tons.
Speed on 4-hour full-speed trial, not less than .	29½ knots.
Penalty weight of machinery, including water, .	272.5 tons.

TRIAL REQUIREMENTS.

(*a.*) Progressive trial over a measured-mile course for standardizing the screws.

(*b.*) A full-speed trial of four (4) hours' duration in the open sea in deep water.

(*c.*) An endurance and fuel-oil and water-consumption trial of 12 hours' duration in the open sea in deep water at a speed of not less than an average of 25 knots an hour.

(*d.*) An endurance and fuel-oil and water-consumption trial of 24 hours' duration (this was afterwards changed to 12 hours' duration) in the open sea at an average speed of about 16 knots.

The contract provided that the water consumption, including water used in all auxiliaries in use on the trial, except the water used and made in the evaporating and distilling plant, should not exceed on the full-power trial 15.5 pounds, on the 25-knot trial 17 pounds, and on the 16-knot trial, 25 pounds of water per hour per shaft horsepower of the main propelling turbines.

GENERAL DESCRIPTION OF THE VESSELS.

The hulls are of steel, the structural work is generally galvanized below the berth deck, except work wholly in the fuel-oil tanks. The shear strake is of 18-pound plate amidship and 8 pounds at the ends. The garboard strake is of 20-

pound plate amidship, tapered to 10 pounds at the ends. The main-deck stringer plate is of 18-pound plate amidships, tapered to 8 pounds at the ends.

The main deck runs the full length of the vessel, and a raised forecastle deck runs about one-quarter of the length from the bow. The berth deck extends forward and aft of the machinery space. On the forecastle deck are located the anchors and anchor davits, capstan, a 3-inch semi-automatic gun, pilot house with bridge above and two .30-caliber automatic guns. The steering engine is located in the pilot house. The space on the main deck under the forecastle contains the officers' quarters, galley, crew's washroom, lamp room and store room. On the main deck just abaft the forecastle two 3-inch semi-automatic guns are placed, one on each side; abaft these are stowed four small boats.

On the main deck amidships are two 45-cm. torpedo tubes, one on each side; aft of the engine hatch is a 3-inch semi-automatic gun on the centerline of the vessel, and a small deck house containing a crew's washroom, wireless telegraph room and companionway to the berth deck; abaft this deck house is placed a 45-cm. torpedo tube on the centerline of the vessel. The torpedoes for the three tubes are to be stowed in the tubes. On the main deck aft is one 3-inch semi-automatic gun on the centerline, the hand-steering gear and the steering quadrant. On the berth deck forward are stores, paint and oil room and quarters for the petty officers and the crew. On the berth deck aft are stores and quarters for the chief petty officers and crew. Below the berth deck forward are the trimming tanks, chain lockers, store rooms and fresh-water tanks, magazine, handling room and fuel-oil tank. Abaft the berth deck forward are fuel-oil tanks, which are separated from the boiler space by a cofferdam. Below the berth deck aft are additional fuel-oil tanks, magazine, handling room and trimming tanks.

There are two boiler rooms with small oil-settling tanks outboard below. There is one engine room the full width of

the vessel. Reserve feed water is carried in a tank under the working platform at the forward end of the engine room.

PARTICULARS OF HULL.

Length between perpendiculars, feet and inches	289–00
over all, feet and inches	293–10½
on load water line, feet and inches	289–00
of straight keel, feet and inches	240–09
Depth, molded, main deck at side, M. S. No. 82½, ft. and in	16–01
Draught, designed above bottom of keel, feet and inches	8–04
Ratio of length to beam at L.W.L.	10.959
Displacement, normal, 4-inch draught, tons	742
per inch at 8-foot 4-inch draught, tons	11.95
Area of immersed midship section, square feet	143
L.W.L. plane, square feet	5,020
wetted surface, square feet	7,700
C. G. of L.W.L. plane, aft of M. S. No. 82½, feet	2.4
C. B. above bottom of keel, feet	5.3
forward of M. S., feet	.13
Transverse metacenter above C. B., feet	7.66
Longitudinal metacenter above C. B., feet	792
Coefficient of fineness, block	.41
midship section	.66
L.W.L.	.66
cylindrical	.63
Number of frames, spaced 21 inches throughout	165
watertight compartments	43
Heights above 8 ft. 4 in. W.L.:	
Top of foremast, feet and inches	59–10½
mainmast, feet and inches	60–03
Bridge (at center, frame 37, top of beam), feet and inches	23–07½
Top of forecastle deck at stem, feet and inches	17–00 3/16
Height of bridge at center above forecastle deck, feet and inches	7–06

CAPACITIES OF WATER TANKS.

	Gallons.	Tons S.W.	Tons F.W.
Compartment A6, fresh water		3,260	12.11
C2, reserve feed		3,867	14.36
	Gallons.	Tons S.W.	Tons F.W.
A1, trimming	430	1.64	1.60
A2, trimming	2,676	10.22	9.94
D9, trimming	2,200	8.41	8.17
D10, trimming	1,216	4.65	4.52
Totals	6,522	24.92	24.23

SMALL BOATS.

Two 20-foot whale boats.
One 20-foot power dory.
One 17-foot dory.

BATTERY.

Five 3-inch (14-pounders) semi-automatic guns.
Two .30-caliber Colt automatic guns.
Three 5-m. by 45-cm. torpedo tubes (twin).

COMPLEMENT.

Commanding officer,	1
Wardroom officers,	3
Men,	85
Total,	89

MACHINERY.

MAIN TURBINES.

There are five ahead turbines and two backing turbines of the Parsons type, arranged on three shafts. The main high-pressure turbine is on the center shaft; the intermediate cruising turbine, one low-pressure and one backing turbine on the starboard shaft. The high-pressure cruising, one low-pressure and one backing turbine are on the port shaft. The turbines were designed to run at 775 r.p.m. at the speed of 29½ knots and at that speed to develop about 12,000 shaft horsepower.

The turbines are of the usual Parsons type. The casings are of cast iron, the rotors have forged-steel drums and forged-steel spindles. The blading is of rolled composition held in place by the usual packing pieces, which are caulked in to the blading grooves. The blading on the three low-pressure stages is of the Parsons wire-root type, made up in sections. The turbine particulars are tabulated below:

TURBINE DATA.

Expansion.	.....	Ahead.				Astern.
		H.P.C.	I.P.C.	M.H.P.	L.P.	
1	Diameter of cylinder, inches	30⅜	30⅜	32¾	46¾	31½
1	Length of cylinder, this diameter, inches	16½	16⅜	13½	7	9⅜
1	Rows of blading	18	16	10	5	8
1	Length of blades, inches	0⅜	0½	1⅜	1⅜	0½
1	Thickness of caulking sections, c.	120	120	130	130	120
2	Diameter of cylinder, inches	31	31½	34	48	32½
2	Length of cylinder, this diameter, inches	16½	20⅜	13⅜	7½	11½
2	Rows of blading	18	16	10	5	8
2	Length of blades, inches	0½	1½	2½	2½	1⅜
2	Thickness of caulking sections, c.	120	130	130	130	130
3	Diameter of cylinder, inches	31⅜	32	35½	49½	35
3	Length of cylinder, this diameter, inches	17⅜	20½	17	7⅜	14⅜
3	Rows of blading	18	16	10	5	8
3	Length of blades, inches	0 1/16	1½	3	3	2⅜
3	Thickness of caulking sections, c.	120	130	240	130	240
4	Diameter of cylinder, inches	...	...	38	52	35
4	Length of cylinder, this diameter, inches	...	...	18½	8½	15
4	Rows of blading	...	...	10	5	8
4	Length of blades, inches	...	...	4½	4½	2⅜
4	Thickness of caulking sections, c.	...	...	240	130	242
5	Diameter of cylinder, inches	...	...	...	54	...
5	Length of cylinder, this diameter, inches	...	...	...	8	...
5	Rows of blading	...	...	...	4	...
5	Length of blades, inches	...	...	...	5½	...
5	Thickness of caulking sections, c.	...	...	...	240	...
6	Diameter of cylinder, inches	...	...	...	59	...
6	Length of cylinder, this diameter, inches	...	...	...	8½	...
6	Rows of blading	...	...	...	4	...
6	Length of blades, inches	...	...	...	7⅜	...
6	Thickness of caulking sections, c.	...	...	...	240	...
7	Diameter of cylinder, inches	...	...	...	59	...
7	Length of cylinder, this diameter, inches	...	...	...	8	...
7	Rows of blading	...	...	...	4	...
7	Length of blades, inches	...	...	...	7⅜	...
7	Thickness of caulking sections, c.	...	...	...	241	...
8	Diameter of cylinder, inches	...	...	...	59	...
8	Length of cylinder, this diameter, inches	...	...	...	8	...
8	Rows of blading	...	...	...	4	...
8	Length of blades, inches	...	...	...	7⅜	...
8	Thickness of caulking sections, c.	...	...	...	242	...

Diameter of rotor drums: H.P.C., 30 inches; I.P.C., 29 inches; M.H.P., 29½ inches; L.P., 43½ inches.

The turbine drains and turbine relief valves and turbine details are generally similar to corresponding parts on torpedo-boat destroyers *Smith* and *Lamson*. The particulars of the shafting are as follows:

	Center.	Wing.
Line Shafts:		
Number of sections	4	2
Diameter outside, inches	6¼	6¼
of hole, inches	3½	3½
coupling flange, inches	12	12
Thickness of coupling flange, inches	1½	1½
Number of taper coupling bolts	8	8
Diameter of bolts at face of coupling, inches	1½	1½
Total length line shafting, feet and inches	67–1½	23–2
Stern-tube Shafts:		
Diameter outside, inches	...	6¼
of hole, inches	...	3½
Length, feet and inches	...	20–7
Thickness of brass sleeves at bearing, inches	...	$0\frac{7}{16}$
Propeller Shaft:		
External diameter, inches	6¼	6¼
Diameter of hole, inches	3½	3½
Length, feet and inches	18–11	18–9½
Thickness of brass sleeves at bearings, inches	$0\frac{7}{16}$	$0\frac{7}{16}$
Length of shafting from flange at after end of rotor shaft, feet and inches	85–11½	62–6½

The centerline shaft has five steady bearings and two stern-tube bearings; each wing line of shafting has two steady bearings, two stern-tube bearings and one strut bearing. The steady bearing and the caps are cast iron, white-metal lined and fitted for forced lubrication. The stern-tube and strut bearings are of composition of the usual Naval type lined with lignum vitæ.

The propellers are solid monel-metal castings. The faces of the blades are machined to true screw surfaces. The backs of the blades are trued to the drawing and the entire wheel is polished.

Propeller Data:	
Number of blades, each	3
Diameter, feet and inches	5–03
Pitch, feet and inches	4–10
Projected area, square feet	12.98
Helicoidal area, square feet	14.10
Disc area, square feet	21.648
Ratio, pitch to diameter	.921
projected area to disc area	.6
Thickness of blade at root, inches	2½
Immersion of upper tip of blade at 8 feet 4 inches draught:	
Wing, inches	36½
Center, inches	53

MAIN CONDENSERS.

There are two main condensers of semi-circular top and bottom sections joined by flat sides. The shells are of steel plate stiffened by angles and ties. The heads are made up of composition frames and sheet copper, the tube sheets are rolled naval brass and the tubes of composition, Admiralty mixture. The tubes are packed with cotton-tape packing and screwed ferrules, and are supported by supporting plates and brass ferrules.

Particulars of One Condenser:

Length between tube sheets, feet and inches	14–0
Height of shell, inside, feet and inches	7–1
Width of shell (mean), feet and inches	2–6
Thickness of shell plates, sides, inch	0½
Tube sheets, inch	1
Number of tubes	2,096
Outside diameter of tubes, inch	0⅝
Thickness of tubes B.W.G., number	18
Cooling surface, square feet	4,800
Diameter of circulating water, inlet, inches	16
outlet, inches	16
Area of exhaust inlet, square feet	15.37
Diameter of main air-pump suction, inches	9½
augmenter suction, inches	9

AUXILIARY CONDENSER.

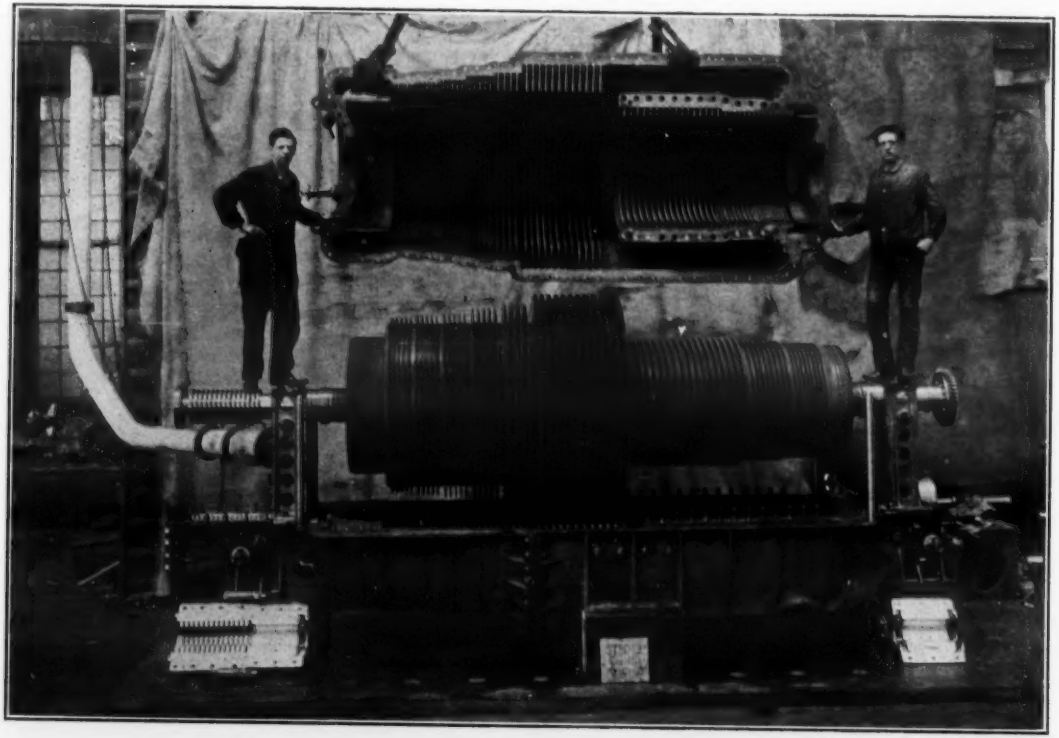
There is one auxiliary condenser mounted on a combined air and circulating pump. This condenser is located aft on starboard side of the engine room.

Particulars of Auxiliary Condenser:

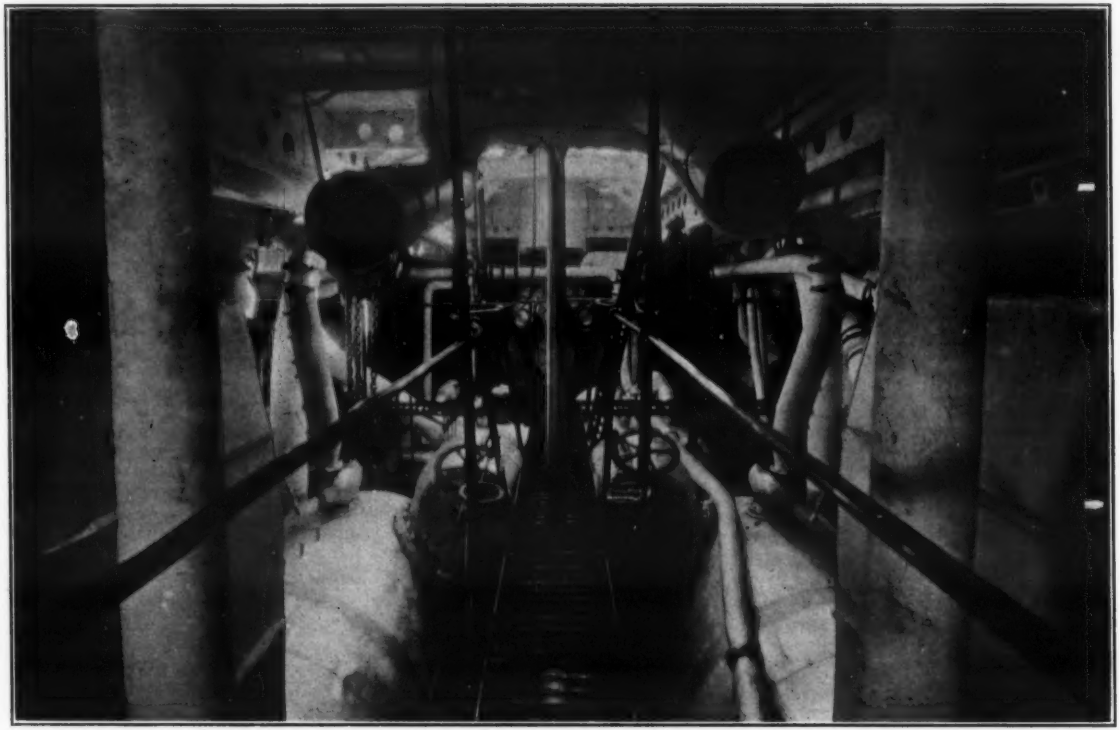
Length between tube sheets, feet and inches	5–1
Number of tubes	302
Outside diameter of tubes, inch	0⅝
Thickness of tubes B.W.G., number	18
Cooling surface, square feet	250
Diameter of auxiliary exhaust inlet, inches	4
air and circulating-pump connection, ins.	4

Auxiliary Air and Circulating Pump:

Diameter of steam cylinder, inches	6
air cylinder, inches	8
circulating-water cylinder, inches	8
Stroke of all pistons, inches	8



LOW-PRESSURE TURBINE, U. S. T. B. DESTROYER "ROE," ON TESTING BLOCKS.



ENGINE ROOM, FORWARD, U. S. T. B. DESTROYER "TERRY."

MAIN AIR PUMPS.

There are two main air pumps of the Davidson vertical twin-beam single-acting type, each having two steam cylinders and two pump cylinders. They are located at the after end of the engine room, near the centerline of the ship and as low as feasible.

Particulars of One Main Air Pump:

Diameter of steam cylinders, inches	10
pump cylinders, inches	22
stroke of all pistons, inches	16
diameter of suction, inches	9½

VACUUM AUGMENTERS.

Two vacuum augmenters and one augmenter condenser of the Parsons type are fitted for increasing the vacuum. The augmenter consists of the usual Parsons steam injectors, which take their suction from the forward ends of the main condensers and discharge to the augmenter condenser and from it by the usual water seal through the main air-pump suction.

MAIN CIRCULATING PUMPS.

There are two double-inlet circulating pumps, each driven by a single-cylinder vertical engine, fitted with forced lubrication. They are located at the outboard sides of the engine room, forward of the main condensers.

Particulars of One Main Circulating Pump:

Diameter of impeller, inches	28
Width of impeller at tip of blade, inches	3½
Diameter of suction nozzle, inches	16½
discharge nozzle, inches	16
steam cylinder, inches	8
Stroke of steam piston, inches	6

FEED-WATER HEATER.

One feed-water heater is installed in the after end of the engine room. The heater has a steel shell with an expansion joint in the center, and dished forged-steel tube sheets. It is fitted with Admiralty-metal composition tubes expanded into the tube sheets. The heads are of cast steel of spherical form.

Particulars of Feed-Water Heater:

Inside diameter of shell, inches	20½
Length between tube sheets, feet and inches	8-0
Diameter of tubes, inches	0⅝
Thickness of tubes, B.W.G., number	18
Tube-heating surface, square feet	500

MAIN FEED TANK.

The main feed tank is located at the after engine-room bulkhead, on the centerline of the ship just abaft the main air pumps. The capacity of this tank amounts to 86.8 cubic feet, and is constructed of galvanized-steel plates and angles, fitted with a bolted steel-plate cover, with the usual air pipes and connections.

BOILERS.

There are four water-tube oil-burning boilers of the Thornycroft type, arranged in two boiler rooms, the forward and after boilers each having its own smokepipe. The two amidship boilers are connected to one common smokepipe. The tubes are seamless-drawn steel. Each boiler has one steam drum and two water drums. The steam drum is 3 feet 6 inches inside diameter, and the water drums are 19 inches inside diameter. The flow of the products of combustion is direct, there being no baffles or passes to cause the gases to travel longitudinally between the tubes.

A deflecting plate is inserted in the uptake space at the steam drum to keep the gases from short-circuiting to the stack. The boiler casing is constructed of galvanized-steel plates and angles and asbestos and magnesia non-conducting material, made up in sections, for that part of the casing outboard of the tubes. The casing at the back end of the boilers is made up of galvanized-steel plates with asbestos and magnesia non-conducting material, and is provided with a 2-inch air space outside of the non-conducting material. The front end casing is constructed with air spaces 15 inches in depth, extending over the entire front of the furnace between the downcomer tubes and the bottom; this casing is fitted with light doors in front above the openings for the oil burners. These doors are arranged so as to be held open the proper amount

for supplying air to the oil burners, and to close automatically in case of the bursting of a boiler tube.

The furnace is lined at the front and back and at the sides up to the water drums with fire-brick tile, $2\frac{1}{2}$ inches thick and 9 inches by 9 inches square. The bottom of the furnace is lined with fire brick, $1\frac{1}{4}$ inches by 9 inches by 9 inches, laid double thickness with broken joints.

Eleven oil burners of the Thornycroft type, built by the contractors, are installed in the lower part of the front of each boiler. Sight holes are placed so as to observe the products of combustion. The steam drum is connected at each end with each water drum by two downcomer pipes, $4\frac{3}{8}$ inches inside diameter by $\frac{5}{16}$ inch thick. These down-comers are expanded into the drums.

Particulars for One Boiler:

Length of steam drum outside heads, feet and inches	11–10$\frac{3}{16}$
Inside diameter of steam drum, feet and inches	3–06
Thickness of tube sheets, inches	0–01$\frac{3}{16}$
Top plate, inch	0–00$\frac{9}{16}$
Length of water drum outside heads, feet and inches	10–09$\frac{9}{16}$
Inside diameter of water drum, inches	0–19
Thickness of water-drum tube sheet, inch	0–00$\frac{9}{16}$
Diameter of two rows of tubes nearest the fire, inches	0–01$\frac{3}{8}$
other tubes, inches	0–01$\frac{1}{2}$
Thickness of tubes for $1\frac{3}{8}$-inch and $1\frac{1}{2}$-inch tubes,	B.W.G. number 10 and 11
Heating surface, one boiler, square feet	4,500.0
Total heating surface, four boilers, square feet	18,000.0
Volume of furnace, cubic feet	423.0
Designed working pressure, pounds per square inch	250.0
Area of single smokepipe, square feet	13.6
double smokepipe, square feet	27.2
Height of smokepipes above base line, feet and inches	37–06
top burners, feet and inches	32–02
Ratio of area of smokepipes to heating surface	.00331

FUEL-OIL HEATERS.

The fuel-oil heater as first designed consisted of a cylindrical steel shell with forged-steel tube plates, steel tubes and forged-steel heads. The oil was arranged to pass through the tubes. Experience before the trials indicated that it would be impossible to make the tubes tight in the tube sheets.

The steel tubes and tube plates were consequently removed and a copper coil substituted in the same shell; this coil was arranged with no joints inside the heater, so as to obviate any chance for leakage into the steam space. Two heaters were installed in each boiler room and were found more than ample to heat the oil to the desired temperature. On the full-power trial of the *Terry* only one fuel-oil heater in each fireroom was used.

Particulars of One Fuel-Oil Heater:

Diameter of shell, inches	0–12
Length of shell, feet and inches	4–00
Inside diameter of copper pipe in coil, inch	0–01
Thickness of pipe in coil	B.W.G. number 12
Amount of heating surface, external surface of coil, square feet	25.36

A triplex oil-pressure hand pump with 1¼-inch plungers of 3½-inch stroke is installed on the starboard side in each fireroom. These pumps are arranged to draw from the fuel-oil settling tanks and discharge to the burners when raising steam. Two special burners are provided for each boiler for raising steam. Small portable copper-coil heaters are arranged to be used in connection with these special burners.

A connection is made with the compressed-air piping for running the fuel-oil service pumps when getting up steam.

Oil burners of the general type of the Thornycroft burner are not well adapted for use under natural draft, as it is difficult to supply sufficient air for combustion. The builders therefore purchased two Ingram steam-or-air atomizing burners, which could be quickly installed in place of the Thornycroft burners. These burners were frequently used for raising steam and running the auxiliary machinery when at the dock. These burners gave excellent service and could easily be operated without smoking.

FUEL-OIL MEASURING ARRANGEMENT.

The fuel oil was pumped from the bunker tank A10 by the fuel-oil tank-supply pumps to the fuel-oil settling tanks abreast the boilers, as shown in the diagram.

The settling tanks, which are designated as B3 and B4 for the forward fireroom, and B5 and B6 for the after fireroom, were operated separately for each fireroom. Before the trials these tanks were calibrated by filling. Scales were made from these calibrations and placed alongside the gage glasses, which were fitted to each tank in each fireroom near the middle of the fore-and-aft length. During the trials oil was used from opposite sides of the ship, in the two firerooms, so that while oil was being used from the port tank in the forward fireroom, it was being used from the starboard tank in the after fireroom; this arrangement was made in order to avoid listing the ship. The oil used in each fireroom was determined by the gage-glass readings on the settling tanks. The settling tank in each fireroom while not supplying oil to the burners was filled, and before being put in use the quantity of oil was noted. The changing over from one tank to another was done without interruption of flow, and in a very few seconds. During the trials, which required two firerooms, the burners in each fireroom were set at the opening to give the approximate quantity of steam required. When the conditions became settled the burners in the forward fireroom were left set at the required opening and the regulation of steam made by adjusting the burners in the after fireroom, thus leaving the control of the pressure and the steam supply in the hands of the engineers in this fireroom. This arrangement involved a minimum of labor and enabled the steam pressure to be maintained more steadily than could otherwise have been done.

FORCED-DRAFT SYSTEM.

The forced draft is of the closed-fireroom system. There are two fans in each fireroom.

The fans were furnished by the Terry Steam Turbine Company, and are of the Sirocco type, built by the American Blower Company. Each fan is direct connected by a vertical shaft to a Terry steam turbine. The fans are 30 inches diameter by 15½ inches width of runner, and were designed

to furnish 23,000 cubic feet of air per minute, against a pressure of 5 inches in the fireroom when running at 1,400 r.p.m. The bearings are oiled by a system of forced lubrication, which is self-contained in the unit. Each fan engine is provided with an emergency trip valve to stop the engine in case of accident causing excessive speed.

TORSION METERS.

The shaft horsepower was calculated from readings of flashlight torsion meters. These are of the Bevis and Gibson pattern, furnished by the Chadburn Ship Telegraph Company. The discs for these torsion meters were placed at a distance of 22 feet 3 inches apart for the center shaft, and 22 feet 10 inches apart for the outboard shafts. The forward discs were placed in the engine room just abaft the turbines and the after discs in the ammunition-handling room abaft the fuel-oil bunkers D1 and D2. A steel tube was run through the fuel-oil bunkers to provide for carrying the flash from the forward disc to the after disc. This long length of shafting enabled very good readings to be taken, and the operation of the torsion meters was quite satisfactory.

LUBRICATION.

The lubrication of the main-turbine bearings and line-shaft bearings, including thrust bearings, is accomplished by oil forced through the bearings by two pumps located in the after end of the engine room. The oil passes through a Schutte-Koerting film cooler on its way to the bearings and is collected from the bearings by a system of pipes which lead into a tank of 160 gallons' capacity, located as low as possible at the aft end of the engine room. Thermometers are located at the pump on the discharge to the oil cooler, on the discharge after passing the cooler and on the discharge from each of the turbine bearings. The line-shaft bearings drain to an auxiliary tank aft, from which a small pump draws the oil and discharges it to the main-drain oil tank located in the

after end of the engine room. The circulating water for the oil cooler is taken from a pipe connection from the starboard main-circulating pump and from a special circulating steam pump located in the after end of the engine room. The two lubricating-oil pumps are of the Davidson make, vertical-piston, double-acting, simplex type, with 4½-inch steam cylinders, 5-inch oil cylinders, 8-inch stroke. The small oil-drain pump from the auxiliary-drain tank aft is of Davidson make, of the vertical, double-acting, simplex type, with 3½-inch steam cylinder, 4-inch oil cylinder by 4-inch stroke.

The oil-cooler circulating pump is of the Davidson make, vertical-piston, double-acting, simplex type, with 4½-inch steam cylinder, 5-inch water cylinder, 6-inch stroke. A reserve-oil tank of 200 gallons' capacity is located in the after end of the engine room.

EVAPORATING AND DISTILLING PLANT.

There are two evaporators and two distillers with their accessories located in the forward starboard side of the engine room. The evaporators have a total capacity of 3,000 gallons of water per 24 hours. The distillers have a total capacity of 2,000 gallons of potable water per 24 hours. The evaporators are of the vertical type with bent U-type coils, each having 75 square feet of tube-heating surface. The coils are of brass tubes 1½-inch outside diameter, No. 13 B.W.G. thick, with ends expanded in the tube sheet of the steam head. The distiller shells are of brass, provided with an expansion joint. The tubes are ¾-inch outside diameter, No. 17 B.W.G. thick, expanded and sweated into the tube sheets. The tube-cooling surface of each distiller is 16.5 square feet. There is one evaporator-feed pump of the vertical, double-acting, simplex type, with 3½-inch steam cylinder, 4-inch water cylinder, 4-inch stroke, and one distiller fresh-water pump of the vertical, simplex type, of the same size as the evaporator-feed pump. The distiller circulating water is taken from the engine-room fire-and-bilge pump.

STEAM PUMPS.

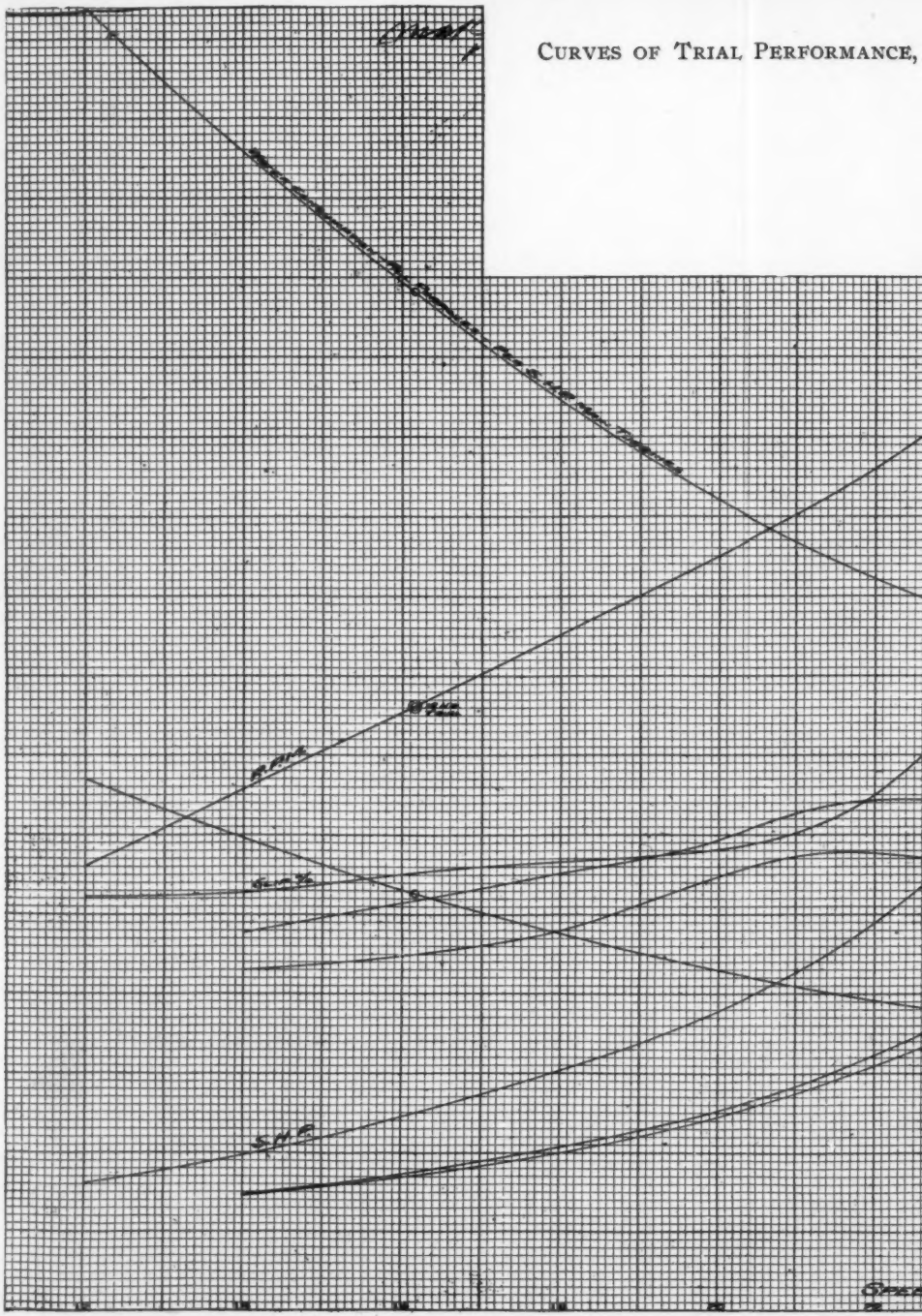
All of the steam pumps in the machinery equipment, excepting the main-condenser circulating pumps, are of the Davidson Company's make. In addition to those previously described there are two main-feed pumps of the vertical-piston double-acting simplex type, with 14-inch steam cylinders, 10-inch water cylinders, 12-inch stroke. These pumps are located at the aft port side of the engine room. There are two auxiliary-feed pumps of the vertical-piston double-acting simplex type, with 14-inch steam cylinders, 10-inch water cylinders by 12-inch stroke. One of these pumps is located on the port side in each fireroom.

There is one engine-room fire-and-bilge pump of the vertical-piston double-acting simplex type, with 10-inch steam-cylinder, 7-inch water cylinder, 8-inch stroke, located on the port side of the engine room forward of the main condenser. There are two fireroom fire-and-bilge pumps of the same type, with 8-inch steam cylinders, 5-inch water cylinders, 12-inch stroke, one located on the port side in each fireroom. There are four fuel-oil service pumps of the vertical-piston double-acting duplex type, with 4½-inch steam cylinders, 2¾-inch oil cylinders, 6-inch stroke. Two of these pumps are located on the starboard side in each fireroom. These pumps draw from the fireroom settling tanks and discharge to the burners. There are two fuel-oil-tank supply pumps, of vertical-piston double-acting simplex type, with 6-inch steam cylinders, 7-inch oil cylinders, 12-inch stroke, one located on the starboard side in each fireroom. These pumps draw from the fuel-oil bunkers forward and aft and discharge to the settling tanks in the firerooms. There is one bilge pump of the vertical-piston double-acting simplex type, with 3½-inch steam cylinder, 4-inch water cylinder, 4-inch stroke, located on the berth deck forward.

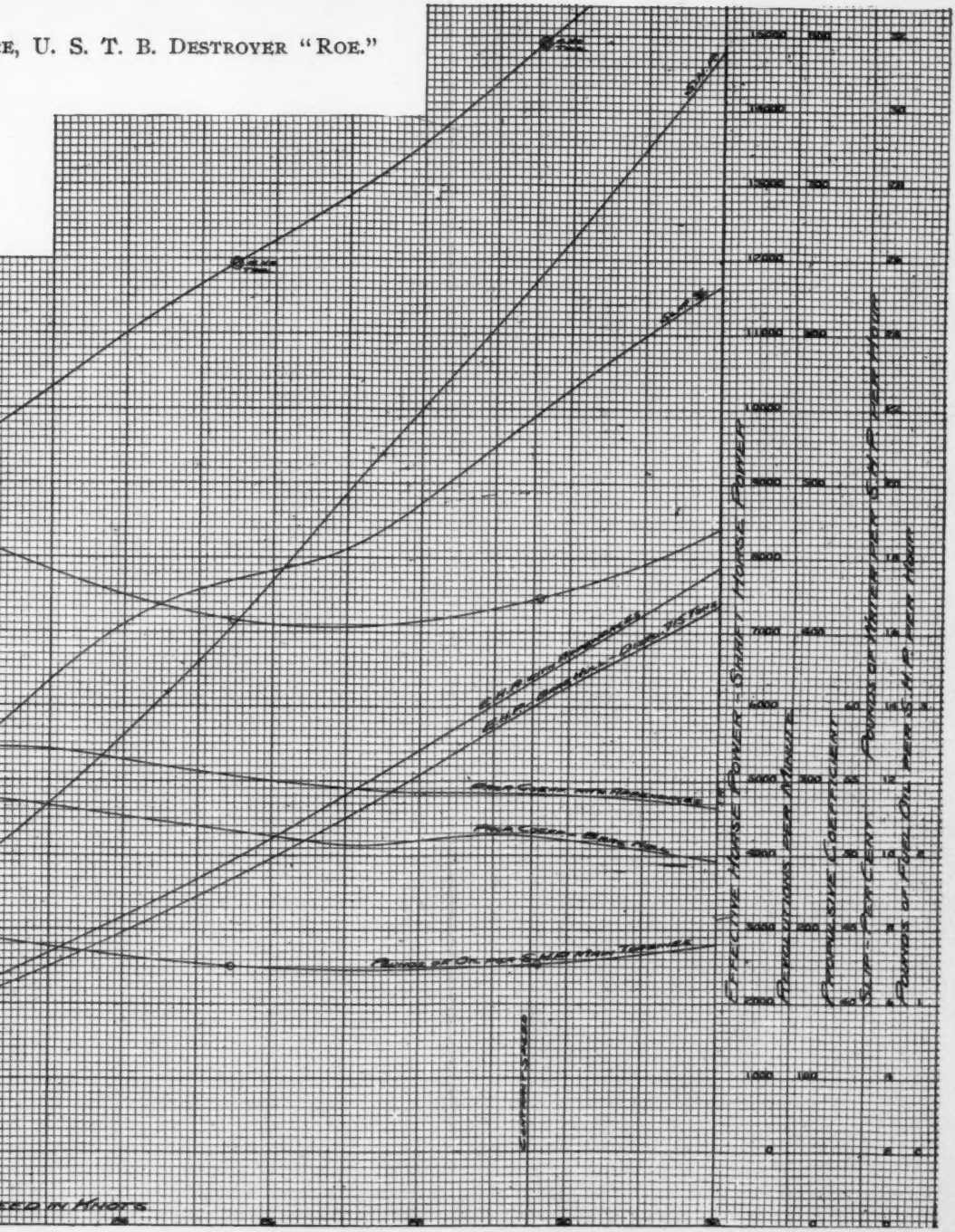
FEED-MEASURING APPARATUS.

During the trials the discharge from the main air pumps was led to calibrated measuring tanks on the main deck, which tanks were filled alternately and discharged by gravity to the

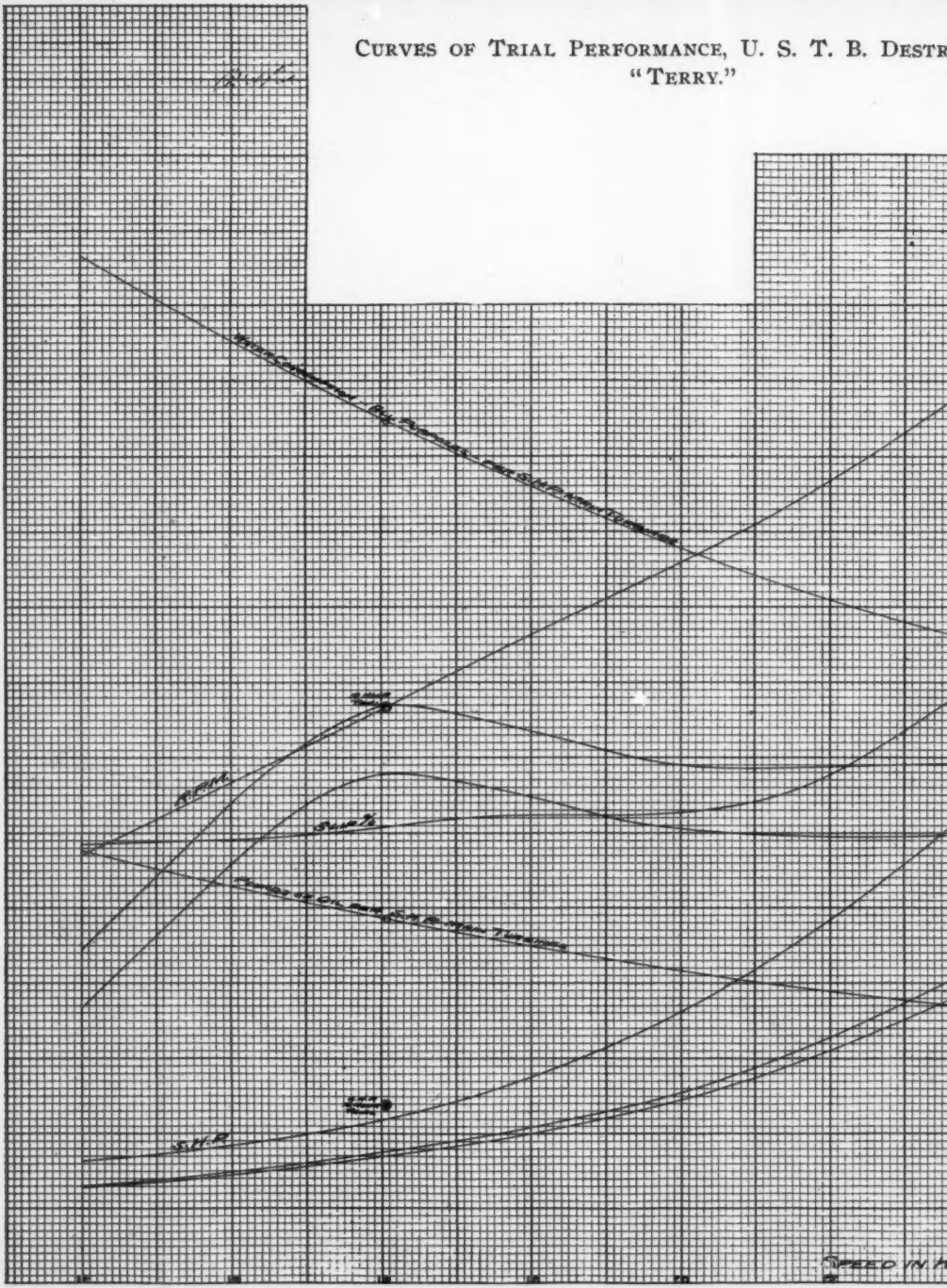
CURVES OF TRIAL PERFORMANCE,

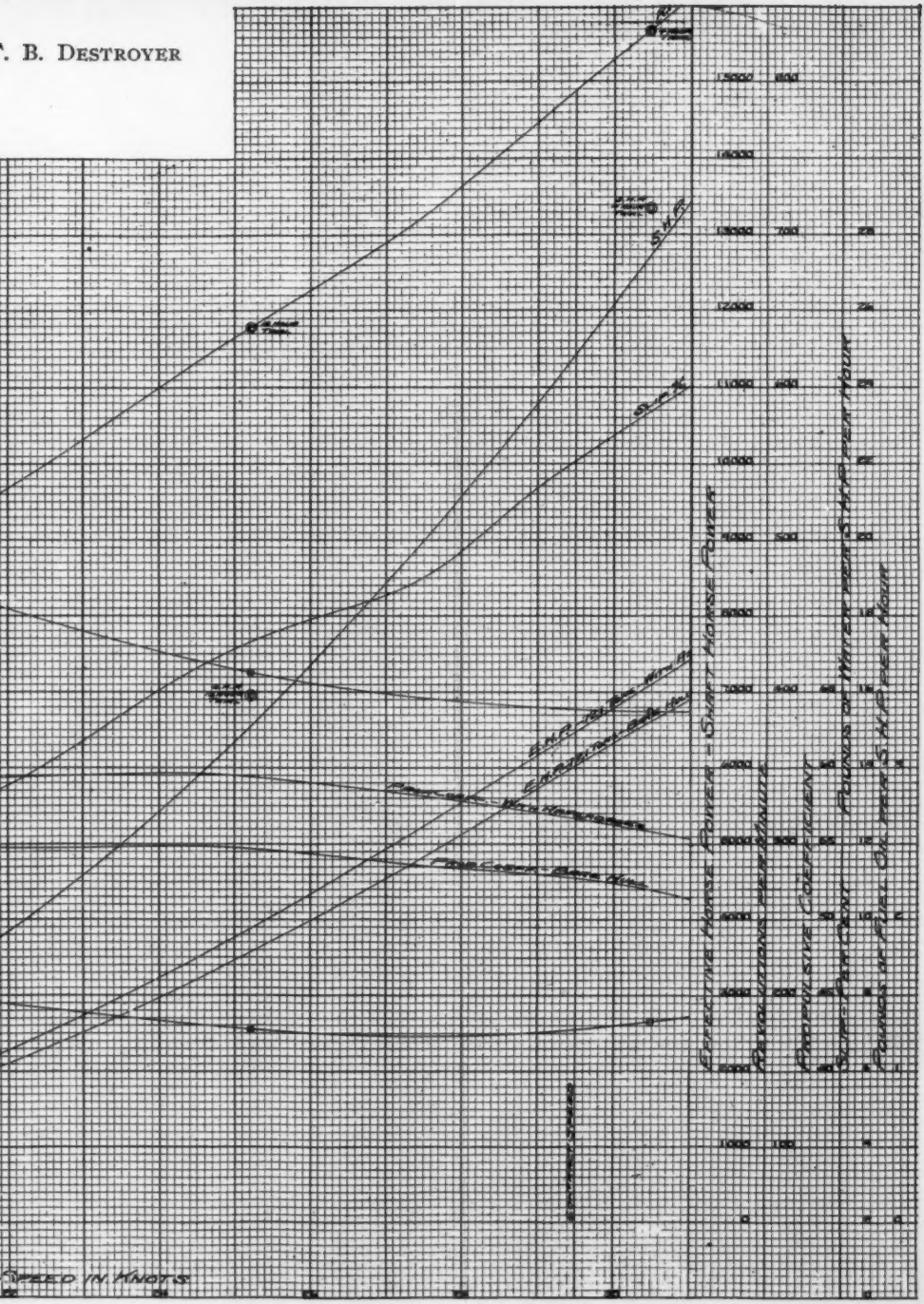


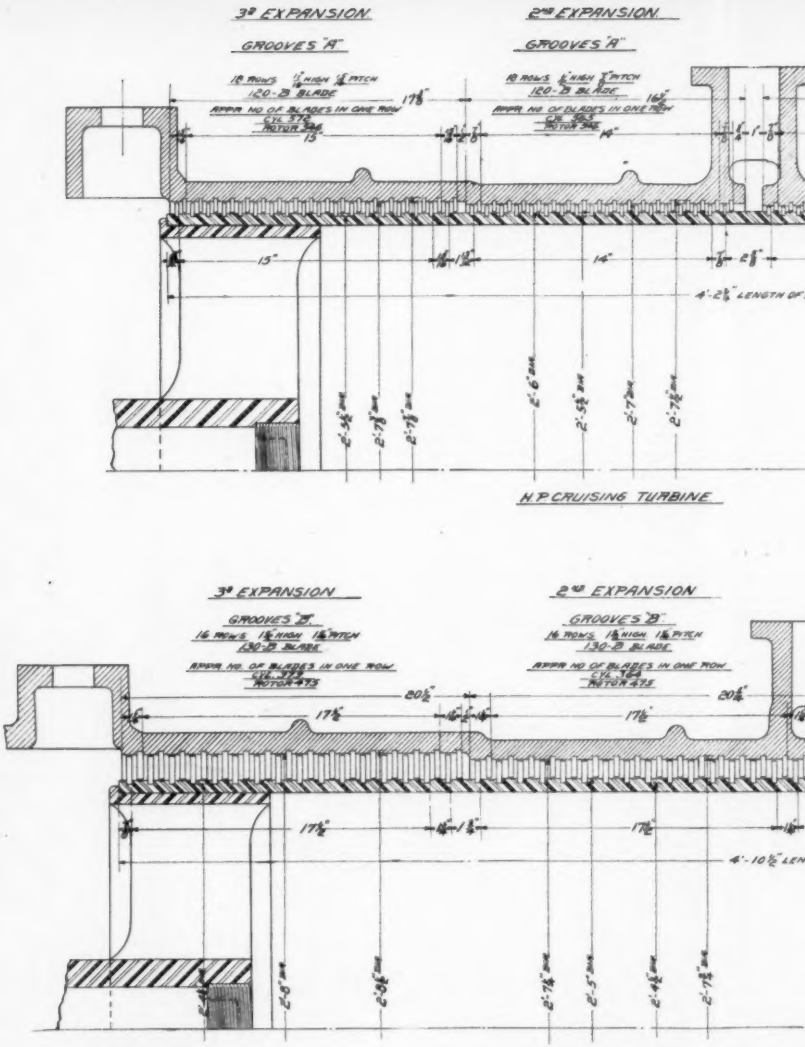
E, U. S. T. B. DESTROYER "ROE."



CURVES OF TRIAL PERFORMANCE, U. S. T. B. DESTR
"TERRY."




Effective Horse Power - Shaft Horse Power
Revolutions per Minute
Coefficient
Pounds of Fuel per Shaft Horse Power per Hour
Pounds of Fuel per Effective Horse Power per Hour
Speed in Knots

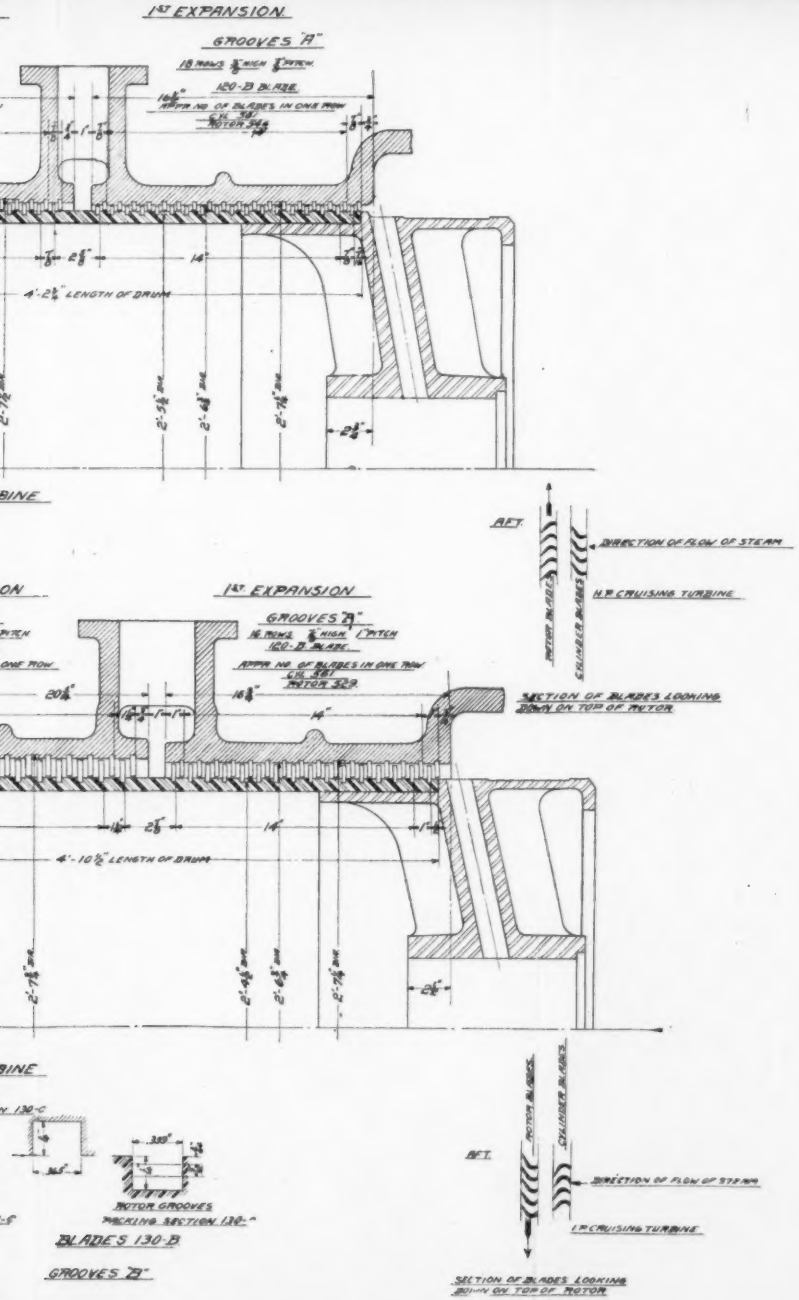


H.P. CRUISING TURBINE		
EXPANSION	TIP CLEARANCE	BLADE SECTION
1st	.03"	120-B
2nd	.03"	120-B
3d	.033"	120-B
I.P. CRUISING TURBINE		
1st	.03"	120-B
2nd	.033"	130-B
3d	.033"	130-B

DUMMY CLEARANCE .015"

FOR H.P. AND I.P. CRUISING TURBINES SEE BLADING LIST DR. NO. 44030

Arrangement of Blading for H.P. and I.P. Cruising Turbin



ING TURBINES, U. S. T. B. DESTROYERS "ROE" AND "TERRY."

feed tank in the engine room. Each tank was of about 1,000 gallons' capacity and fitted with a gage glass and a graduated scale adjacent to the glass. The discharge from the air pumps to the measuring tanks was controlled by a quick-opening, hand-operated valve, as was also the discharge from the measuring tanks to the feed tank.

During the trials the interval of time required for filling each tank was noted continually by a stop watch as a means of checking the tallying of the tanks.

VENTILATION.

The ventilation of the vessel is by natural means. As originally planned the engine room was provided with one ventilator, 36 inches in diameter, located in the forward end of the engine hatch. The engine skylight and a trunk aft were arranged for exhaust of the warm air. The dock trials of the *Roe* and the builder's sea trial of this vessel demonstrated that additional ventilation for the engine room would be necessary. During the official trials of both the *Roe* and *Terry* electric fans were temporarily installed to assist the ventilation. After the trial of the *Roe* two additional 15-inch ventilators were placed at the forward end of the engine room, one on each side of the engine hatch; one 15-inch ventilator was placed at the after end of the engine hatch, and a similar ventilator was located on the port side forward of the trunk at the aft end of the engine room; a 24-inch ventilator was placed on the starboard side of the trunk at the aft end of the engine room. These ventilators were arranged with cowls, and provision was made for unshipping the ventilators and replacing them with watertight covers when necessary.

WINDLASS AND CAPSTAN.

The windlass, which is located on the forecastle deck with engine below deck, is of the Hyde Steam Windlass Company's vertical type, with vertical shaft, driven by worm gear, worked from a worm located on the crank shaft of the engine.

The shaft carries on its upper end and above the forecastle deck a wildcat and gypsy head, with locking gear complete. The wildcat is of open-hearth steel to suit $\frac{7}{8}$-inch close-link chain cable. The windlass engine is of the double-cylinder type, with cylinders 4 inches in diameter by 4 inches stroke, designed for a working steam pressure of 200 pounds per square inch.

STEERING ENGINE.

The steering engine is located in the pilot house on the forecastle deck and is of the combined hand and steam-gear type, built by the Lidgerwood Manufacturing Company. A composition steam-steering column of the Bureau's standard design is located on the top of the pilot house. The steering engine is connected by $\frac{7}{8}$-inch close-link B. B. Crane chain to rods led along the outboard side of the main deck to a cast-steel quadrant aft. Forward of the quadrant aft sliding blocks are fitted and connected by similar chain to a double hand-steering wheel. A compass is located on the pilot house, one in the pilot house and one aft, just forward of the hand-steering wheel.

ELECTRIC PLANT.

There are two 5-kilowatt turbo-generating sets of 125-volt pressure at the terminals; one 1-kilowatt motor generator for wireless outfit; 125 regular electric fixtures; one 18-inch distant hand-controlled searchlight; one electric 2-light night-signal set with keyboard; one portable ventilation set, $\frac{1}{4}$-H.P.; eight $\frac{1}{12}$-H.P. bracket fans; a set of ship's running lights; two cargo reflectors with outlets; a system of interior communication with motor generator, batteries, switches, lights, instruments, etc., and a switchboard with necessary instruments. The generating sets are of the General Electric Company's make, with Curtis steam turbines. The generators and turbines are of the horizontal type. The wiring is of the two-wire feeder system.

AIR-COMPRESSOR PLANT FOR TORPEDOES.

In the forward starboard side of the engine room is located the air-compressor plant for the torpedoes. The air compressor and accumulators are of the Navy type for torpedo-boat destroyer work. In addition to the air connections for the torpedoes a lead is taken into the fireroom. In each fireroom two connections are provided for compressed-air tools and for cleaning boilers. A connection is also made for running the fuel-oil service pumps. Two connections are provided in the engine room for air tools.

STEAM-HEATING SYSTEM.

The steam-heating system is of the usual naval type. The radiators are made up of brass pipe. There are four circuits, as follows:

Circuit No. 1.—For crew and engine-room force and petty officers forward.

Circuit No. 2.—For officers' quarters, crew's washroom and pilot house forward.

Circuit No. 3.—Crew's space and washroom aft.

Circuit No. 4.—Galley.

BUILDER'S TRIALS.

Dock trials of the machinery of the *Roe* were run June 26, 1910. The builder's sea trial was conducted in the Chesapeake Bay, June 30, 1910.

The dock trials of the machinery of the *Terry* were run September 4, 1910. The builder's sea trial was run in the Chesapeake Bay, September 10, 1910.

During the trials of both boats the main and auxiliary machinery was operated under various conditions. The sea trial of the *Roe* was held before the vessel was docked and painted; consequently the results obtained were much affected by foulness of the ship's bottom. The oil coolers for the forced-lubricating system for the turbines did not prove entirely satisfactory owing to the small area for the passage

of oil through the cooler, which required a high pressure to be carried on the discharge from the lubricating-oil pumps. Before the builder's trials of the vessel cocks were installed in the oil-discharge pipes at the bearings; these cocks were intended to regulate the flow of oil to the bearings and were so designed as to prevent the entire cutting off of oil from any bearing.

The fuel-oil heaters on the *Roe* gave trouble owing to the leaking of the tubes. These heaters were altered before the official trials, as noted elsewhere. After the change no further trouble was experienced with them. The main low-pressure thrust bearings gave some slight trouble on the *Terry* by heating. This trouble did not develop on the trial of the *Roe*. These bearings afterwards gave some trouble on the official four-hour trial of the *Roe*, as noted later.

The boilers, blowers, fuel-oil system, engine-room auxiliaries and main turbines all worked efficiently, with the exceptions noted above.

The dock and builder's trials on the *Roe* demonstrated the need of additional ventilation in the engine room, which was later provided for.

OFFICIAL TRIALS.

STANDARDIZATION TRIAL OF THE *ROE*.

On July 12th and 13th the standardization runs were made over the Delaware Breakwater measured course, near Lewes, Delaware. After several runs had been made on July 12, the trial was postponed owing to thick weather. On July 13 the standardization trials were completed. The estimated displacement at the middle of the high-speed runs on July 13 was 707 tons.

STANDARDIZATION TRIAL OF THE *TERRY*.

The standardization trials were run on September 20th and 21st over the Delaware Breakwater course. A few runs were made on September 20, but the ranges were obscured by haze

TURBINE DATA.—STANDARDIZATION TRIALS, U. S. S. "ROE," OFF DELAWARE BREAKWATER, JULY 12 AND 13, 1910.

No. of run.	Time.			Revolutions per minute.				Steam pressures.										Vacuum in inches of mercury.		Temperatures.										Torsion meter readings. K = 2.2438.			Brake horsepower.			
								Engine room, gage.			In steam belt, absolute.									In engine room.	Injection.		Discharge.		Air-pump discharge.		Heaters, feed.		Outside air.							
	H.	M.	S.	Shaft No. 1.	Shaft No. 2.	Shaft No. 3.	Average all shafts.	Main steam.	Auxiliary steam.	Aux. exhaust.	Main H.P. turbine.	Cruising turbines. H.P.	Cruising turbines. I.P.	L.P. turbines. P.	L.P. turbines. S.	On glands, gage.	P.	S.		P.	S.	P.	S.	P.	S.	Feed to.	Feed from.		Shaft No. 1.	Shaft No. 2.	Shaft No. 3.	Shaft No. 1.	Shaft No. 2.	Shaft No. 3.	Total.
1	9	49	44	300.24	263 04	301.75	288.34	225	250	8	15	170	65	10	7.5	...	29	29	116	68	68	79	78	83	83	120	192	78	.425	.235	.380	282.9	138.7	257.2	678.8
2	10	8	58	300 28	260 79	308 40	289.82	235	240	7	15	175	65	7.5	7.5	3	29.5	29	116	70	71	80	80	83	84	118	194	79	.425	.235	.390	289.7	137.5	269.8	697.0
3	10	26	9	326.04	280.02	332.11	312.72	240	250	5	15	175	65	7.5	6.5	3	30.5	29.5	116	72	71	79	81	83	84	119	207	79	.475	.170	.425	343.8	106.8	316.7	767.3
4	10	45	12	364 23	314.38	360 27	346.29	255	240	5	30	240	95	10	10	2.5	29.5	28.5	116	71	71	82	81	85	88	110	170	73	.510	.335	.510	494.4	236.3	412.3	1,143.0
5	11	2	52	381.57	325.88	388.78	365.41	250	250	4	25	260	92	10	8.75	2.5	30.5	30	114	71	70	81	81	84	84	108	174	73	605	.355	.585	599.3	259.6	510.3	1,369.2
6	11	17	45	378.99	318.97	381 87	359.94	250	255	4	32.5	260	100	12.5	11.25	2.5	29	28	116	70	69	80	81	82	84	103	158	74	.605	.325	.585	595.3	232.6	501.3	1,329.2
7	11	40	7	...	...	...	...	253	260	6	60	2.5	205	15	15	1.5	30.5	29.5	116	65	69	86	82	88	83	100	165	73	...	...	...	...	...	...	...
8	1	44	42	512.15	424.56	460.34	465.68	245	260	10	60	2.5	205	19	18	2	29	29.5	108	67	68	82	85	80	82	96	152	68	...	...	...	...	...	...	...
9	1	58	32	465.11	369.60	362.88	399.20	240	260	12	65	2 5	210	26	24	3	28	28.5	119	69	68	88	85	82	82	95	140	68	...	...	...	...	...	...	...
10	2	14	1	485.49	397.02	418.38	433 63	255	255	11	65	7.5	210	20	19	2.5	28.5	29	112	67	67	85	85	81	82	95	140	68	...	...	...	...	...	...	...
11	3	31	58	520.92	426 68	439.52	462.37	250	270	10	70	7.5	225	21	20	3.5	28.5	29	117	67	68	85	88	80	80	92	133	71	1 005	.570	.790	1,168.8	545.8	779.7	2,494.3
12	3	46	38	525.75	431.59	447.85	468.40	250	260	10	70	7.5	225	22	21	3.5	28.5	28.5	119	68	68	85	88	80	83	96	138	73	1.195	.575	.690	1,403.7	556.8	693.4	2,653.9
13	4	1	28	523.08	425 72	437.09	461.96	245	260	10	75	7.5	225	26	25	3 5	29	28.5	120	68	67	85	88	82	83	92	141	75	1.140	.580	.590	1,332.1	554.1	578.7	2,464.9
14	4	18	46	570.37	468.50	487.16	508.68	265	275	12	75	6.25	257	24.5	23	4	29 5	29.5	124	68	68	86	88	86	84	98	141	77	1.445	.790	.850	1,842.8	830.5	929.9	3,603.2
15	4	33	32	564.51	461 94	474.21	500.22	255	275	14	82	5	260	25	24	3	28.5	28.5	123	67	69	86	91	82	84	95	140	77	1.415	.735	.625	1,786.0	761.9	665.7	3,213.6
16	4	47	3	561.04	459.28	467.42	496.11	260	275	10	90	5	270	27	26	1.75	28	29	124	70	69	90	91	85	85	98	150	76	1.430	.720	.585	1,795.8	742 0	613.6	3,151.4
17	5	7	33	...	...	...	...	240	255	8	150	5	250	36	36	5.5	28	28 5	110	70	68	97	90	92	85	101	144	75	...	...	...	...	...	...	...
18	5	21	14	...	...	...	...	250	260	11.5	155	5	255	39	38	4.5	28	28	122	68	68	102	90	102	100	110	160	77	...	...	...	...	...	...	...
19	10	16	9	659.93	595.10	600.14	618.39	260	265	6	110	2.5	265	28	28	3	29.5	29.5	125	72	72	93	92	84	90	96	140	79	1 865	1.390	1.180	2,761.4	1,856.0	1,589.0	6,206.4
20	10	28	34	658.88	600.10	603.8(	620.93	255	265	6	110	2.5	265	28	27	3	29.5	29.5	127	72	72	93	92	83	90	100	155	81	1.845	1.445	1.120	2,727.7	1,945.7	1,517.4	6,190.8
21	10	41	22	658 30	597.26	601.58	619.05	255	265	7	110	2.5	262	27.5	27	3	29	29.5	130	73	72	94	91	95	90	103	164	83	1.850	1.445	1.110	2,732.4	1,936.3	1,498.3	6,167.0
22	10	56	34	611.26	515.36	539.32	555.31	260	275	7	70	2.5	235	21	20	1.5	29	29.5	129	72	73	88	89	85	84	102	190	84	1 675	.960	.940	2,297.8	1,110 3	1,137.8	4.545.9
23	11	9	28	614.56	513.38	542.64	556.86	255	263	10	70	2 5	240	21	20	2	29.5	29.5	132	72	74	88	87	88	84	106	200	84	1.660	.970	.910	2,289.3	1,117 3	1,108.2	4,514.8
24	11	22	51	615.10	515.65	544.73	558.49	252	263	10	70	2 5	242	21	20	2	30	29.5	132	71	72	89	100	86	84	105	203	84	1.730	.965	.905	2,387.8	1,116.4	1,106.3	4,610.5
25	11	47	44	687.56	707.69	699.20	698 15	200	190	12	175	2.5		.5	42	40.5	2.25	26.5	28.5	132	71	72	106	102	90	90	105	182	83	1.810	2.270	1.825	2,792.0	3,604.0	2,862.8	9,258.8
26	12	5	58	707.72	714.83	711.86	711.47	215	220	7	175	2.5		.5	42	39.5	3.25	27 5	30	131	70	73	102	102	98	97	110	190	84	1.830	2.270	1 830	2,906.0	3,641.0	2,923.0	9,470.0
27	12	21	32	702.99	710.48	710.7(	708.07	200	215	15.5	175	2.5		.5	41.5	39.5	3.25	27.5	28	132	74	74	105	104	98	95	98	180	83	1.790	2.265	1 860	2,823.1	3,615.0	2,966.3	9,404.4
28	12	35	59	795.32	795.40	805.82	798.85	220	235	16	215	2.5		.5	50	48	2.75	27.5	28.5	132	72	74	106	104	98	100	118	195	84	2.050	2.535	2.035	3,657.8	4,524.3	3 681.0	11,863.1
29	12	47	11	774.06	776.39	788.41	779.62	222	235	9	210	2.5		.5	48	44.5	2.5	28	28.5	130	70	74	106	104	100	112	182	83	2.000	2.525	2.065	3,473.3	4,398.5	3,653.4	11,525.2
30	12	58	24	790.69	789.63	801.50	793.94	245	250	16	210	2.5		.5	49	45	2.75	28	28.5	130	69	74	105	104	100	105	116	190	82	2.025	2.535	1.990	3,592.5	4,491.5	3,579.0	11,663.0
31	1	14	9	865.32	866.79	873.32	868.48	235	245	19	242.5	2.5		.5	56	52	3.25	27.5	28 5	130	69	73	109	107	110	110	125	200	82	2.225	2.765	2.230	4,320.8	5,377.8	4,370.0	14,068.6
32	1	24	56	870 64	874.27	885 15	876.68	240	257	17	245	2 5		.5	55.5	52	3	27.5	28	128	70	74	110	106	110	112	125	190	82	2.145	2.770	2.200	4,192.5	5,434.8	4,369.8	13,997.1
33	1	35	6	868.83	870.82	882.49	874.05	243	260	14	245	2.5		.5	56	52	3	27	28	128	68	76	108	105	112	112	120	180	82	2.300	2.730	2.240	4,484.5	5,334.3	4,435.8	14,254.6
34	1	45	43	870.05	874.56	887.43	877.35	245	260	13	245	2.5		.5	55.5	52.5	3	27	28	130	70	74	108	106	112	112	120	194	82	2.210	2.725	2.250	4,314.5	5,347.5	4,480.5	14,142.5
35	1	56	16	856.28	860.07	872.67	865.01	227	245	13	240	2.5		.5	55	52	3	27.5	28	131	70	70	108	107	112	112	120	186	82	2.130	2.725	2.220	4,092.5	5,258.3	4,347.0	13,697.8

TURBINE DATA.—STANDARDIZATION TRIALS, U. S. S. "TERRY," OFF DELAWARE BREAKWATER, SEPTEMBER 20–21, 1910.

No. of run.	Revolutions per minute.				Steam pressures.							Vacuum in inches of mercury.		Temperatures.								Torsion meter readings.			Brake horsepower.			
					Engine room, gage.		In steam belt, absolute.																					
	Shaft No. 1.	Shaft No. 2.	Shaft No. 3.	Average all shafts.	Main steam.	Aux. exhaust.	Main H.P. turbine.*	Cruising turbines.		L.P. turbines.		On glands gage.		In engine room.	Injection.		Discharge.		Air-pump discharge.		Heaters, feed.		Shaft No. 1. K = 2.22.	Shaft No. 2. K = 2.244.	Shaft No. 3. K = 2.1835.	Shaft No. 1.	Shaft No. 2.	Shaft No. 3.	Total.
								H.P.	I.P.	P.	S.	P.	S.	P.	P.	S.	P.	S.	P.	S.	Feed to.	Feed from.							
1	318.165	296.32	318.335	310.94	235	23	20	27	20	7.5	7.5	*29.5	*29.5	90	70	70	74	74	75	75	86	180	.34	.357	.333	240	237	231	708
2	319 45	297.18	319.22	311.95	225	21	20	27	20	7.5	7.5	1	29 5	29.5	92	70	70	74	74	74	74	88	125	.34	.457	.318	241	305	222	768
3	309.06	288.21	309 48	302.75	225	21	20	27	20	7.5	7.5	1	29.5	29.5	90	70	70	76	76	75	76	89	115	.34	.380	.312	233	246	211	690
4	438.43	319.12	443.76	400.44	220	15	25	180	60	12.5	12.5	2.5	29.5	29.5	98	70	70	78	78	80	75	91	98	.71	.187	.798	691	134	773	1,598
5	444.33	319.35	448.29	408.99	235	17	25	180	65	12.5	12.5	2.5	29.5	29.5	90	70	70	80	80	84	84	90	96	.73	.167	.783	720	120	766	1,606
6	...	...	...	...	240	17	25	180	65	12.5	12.5	2 5	29.5	29.5	104	70	70	80	80	80	82	91	97	...	...	...	...	...	...	...
7	...	...	...	408.15	230	17	25	180	65	12.5	12.5	2.5	29.5	29.5	96	70	70	79	79	80	82	90	98	...	...	...	...	...	...	...
8	...	...	...	...	245	19	30	140	55	6.5	6.5	1	29.5	29 5	98	70	70	79	79	80	82	86	160	...	...	...	...	...	...	...
9	...	...	...	395.77	245	19	30	135	55	6.5	6.5	1.5	29.5	29.5	105	70	70	79	79	80	82	89	116	...	...	...	...	...	...	...
10	391.18	335.20	393.88	373.42	240	15	25	135	50	9	9	1.5	29,5	29.5	90	69	69	78	78	80	80	86	90	.48	.272	.553	417	205	476	1,098
11	394.65	342.04	399.47	378 72	230	17	25	135	50	9	9	2	29.0	29.5	98	69	69	78	78	80	80	86	88	.48	.382	.533	421	293	465	1,179
12	402.98	348.22	406 84	386.01	225	17	25	135	50	9	9	2	29 5	29.5	96	69	69	78	78	80	80	88	90	.48	.357	.518	429	279	460	1,168
13	547.22	448.28	495.78	497.09	225	19	55	...	165	17	16	3	29.5	29.5	102	69	69	82	82	82	82	92	102	1.23	.652	.818	1,494	656	885	3,035
14	549.29	451.15	496.17	498.85	225	19	55	...	165	17	16	2.5	29.5	29.5	92	70	70	82	82	82	82	90	146	1.23	.631	.828	1,500	639	897	3,036
15	549.27	450.14	496.90	498.77	230	18	55	...	165	17	16	2.5	†28.5	†28.5	98	69	69	82	82	82	82	89	152	1.22	.641	.793	1,488	647	860	2,995
16	632.02	526.55	563.70	574.09	235	18	80	...	240	23	22	2	28.2	28.5	92	69	69	86	86	86	87	92	168	1.71	.972	.978	2,395	1,148	1,204	4,747
17	634.27	527.74	566.75	576.25	250	19	80	...	240	23	22	2	28.5	28.5	100	69	69	86	86	86	87	93	165	1.715	.979	1.013	2,414	1,159	1,254	4,828
18	633.08	526.92	563.53	574.51	255	19	80	...	240	23	22	2	28.2	28.5	98	69	69	84	84	86	87	94	174	1.71	.982	1.008	2,403	1,161	1,240	4,804
19	681.43	598.65	608.33	629.54	250	16	110	...	255	26	26	5	28.5	28.5	100	69	69	89	89	88	88	94	152	1.895	1.337	1.133	2,866	1,796	1,507	6,169
20	677.69	595.02	603.42	625.38	250	15	110	...	260	27	26	3.5	28.2	28.5	98	69	69	90	90	88	89	95	138	1.9	1.332	1.153	2,858	1,779	1,519	6,156
21	686.18	600.97	611.45	632.87	250	16	110	...	260	27	26	4	28.2	28 5	100	69	69	90	90	90	90	104	132	1.92	1.377	1.173	2,924	1,857	1,566	6,347
22	717.41	709.14	717.56	714.70	220	14	175	...	...	41	42	2	28	28.2	98	69	69	96	96	94	94	98	128	1.81	2.162	1.913	2,882	3,441	2,997	9,320
23	725.97	714.17	729.56	723.23	250	14	175	...	...	41	42	1.5	28	28	100	69	69	96	96	94	94	96	140	1.83	2.164	1.883	2,950	3,468	3,000	9,418
24	...	...	...	...	250	14	175	...	...	41	42	3.5	27.5	27.7	98	69	69	96	96	94	94	97	140	...	...	...	...	...	...	...
25	719.25	710.88	723.68	717.94	215	14	175	...	...	40	42	3	27.5	27.5	90	69	69	96	96	92	93	98	138	1.77	2.142	1.868	2,826	3,417	2,952	9,195
20	...	...	...	...	245	15	175	...	...	40	42	3	27.5	28	96	70	70	95	96	93	94	100	150	...	...	...	...	...	...	...
27	726.18	715.35	731.39	...	250	14	175	...	...	40	42	3	27.5	28	92	70	70	96	96	92	93	100	154	1.78	2.164	1.903	2,869	3,473	3,039	9,381
28	721.75	712.35	729.57	721.22	250	14	175	...	...	40	42	3	27.5	28	96	70	70	96	96	94	94	102	158	1.785	2.137	1.878	2,860	3,416	2,992	9,268
29	723.03	713.42	730.63	722.36	250	14	175	...	...	40	42	3.5	27.5	28	90	70	70	96	96	94	94	100	158	1.78	2.160	1.873	2,857	3,458	2,988	9,303
30	793.31	776.35	809.99	793.22	200	13	205	...	...	48	49	3	27	27	98	70	70	102	101	95	95	101	154	2.065	2.437	2.033	3,636	4,245	3,595	11,476
31	810.27	785.10	822 16	805.84	246	14	210	...	...	50	50	2	27	27.5	90	70	70	100	101	95	96	104	156	2.08	2.455	2.068	3,741	4,325	3,713	11,779
32	808.18	782 00	821.86	804.01	250	14	210	...	...	50	50	2	27	27.5	98	70	70	99	101	95	95	104	138	2.09	2.452	2.043	3,749	4,302	3,666	11,717
33	857.25	829 85	871.73	852.94	240	12	230	...	...	55	55	3	26.5	27	90	70	70	102	102	100	100	106	158	2.21	2.617	2.143	4,206	4,873	4,078	13,157
34	811.88	791.51	830.03	811.14	255	14	215	...	...	50	50	2	27	27.5	100	70	70	100	104	101	102	104	160	2.13	2.506	2.108	3,839	4,453	3,821	12,113
35	815 97	803.79	856.30	825.35	245	14	220	...	...	54	50	3	26.5	27	92	70	70	102	100	101	102	104	163	2.12	2.512	2.143	3,840	4,530	4,006	12,376
36	809 90	799.75	859.43	823.05	245	14	220	...	...	54	50	4	26.5	27	98	70	70	101	100	101	101	103	162	2.06	2.487	2.183	3,703	4,463	4,097	12,263
37	804.81	798.35	861.28	821.48	245	14	225	...	...	57	53	4.5	26 5	27	92	70	70	102	100	102	102	103	164	2.15	2.600	2.313	3,841	4,658	4,349	12,848

* Spring gage. † Mercury column.

and the trial was not completed until the next day. The estimated displacement at the middle of the high-speed runs on September 21 was 721 tons.

On the standardization runs of both ships the same combination of turbines was used for corresponding speeds as used on the four-hour full-speed trial; the twelve-hour twenty-five-knot trial and twelve-hour sixteen-knot trial. The speed and revolution curves were constructed from the data obtained during the standardization trials. From these curves it was deduced that the following means of the revolutions of the three shafts would be required for the following speeds:

		Roe.	*Terry.*
16	knots, r.p.m., . . .	377	382
25	knots, r.p.m., . . .	632.5	632.5
29.5	knots, r.p.m., . . .	791	793.5

FOUR-HOUR FULL-SPEED TRIALS.

ROE.

The four-hour full-speed trial was begun about forty-two miles to the eastward of Cape Henry, at 10:50 A. M., July 16, 1910, and was finished at 2:50 P. M., at about the same place. The weather was clear, warm and pleasant. The sea was smooth, with a light ground swell, and there was a gentle breeze from S.W. by S. freshening to a moderate to stiff breeze from S.S.W. at 12:30 P. M.

The mean displacement during this trial was 711 tons.

TERRY.

The four-hour full-speed trial was commenced about 45 miles to the eastward of Cape Henry at 8:55 A. M., September 24, 1910, and was finished four hours later at the same place. The weather was clear and pleasant. The sea was smooth, with light easterly swell and light, variable airs.

The mean displacement during the *Terry's* four-hour trial was 722.5 tons.

In the following data the shaft horsepower of the turbines is taken from the curve of standardization horsepowers corresponding to the average number of revolutions on the trials of the *Roe*. On the *Terry* the shaft horsepower of the turbines is taken as the power obtained on the trials.

PERFORMANCE FOUR-HOUR FULL-SPEED TRIALS.

	Roe.	*Terry*.
Average steam pressures:		
At boiler, gage, pounds	239.0	250.0
Engine room, main steam, gage, pounds	233.0	250.0
auxiliary steam, gage, pounds	237.0	...
Auxiliary exhaust, gage, pounds	13.0	14.0
On glands, gage, pounds	3.6	2.8
Main H.P. turbine, absolute, pounds	217.4	229.0
H.P. cruising turbines, absolute, pounds	2.5	5.2
I.P. cruising turbine, absolute, pounds	0.5	11.3
L.P. turbine, starboard, absolute, pounds	47.0	55.0
port, absolute, pounds	52.2	54.0
Pressure in oil system	...	5.1
Vacuum in condenser, starboard, inches of mercury	27.5	27.5
Vacuum in condenser, port, inches of mercury	27.5	27.0
Air pressure in fireroom, forward, inches of water.	3.42	2.9
aft, inches of water	3.11	3.9
Barometer	29.99	...
Revolutions, or double strokes, per minute:		
Shaft No. 1, starboard	782.85	839.22
No. 2, center	790.00	811.03
No. 3, port	811.87	851.74
Average	794.9	833.99
Pumps, main-air, starboard	56.0	44.8
port	46.7	43.6
circulating, starboard	323.0	376.5
port	347.0	360.0
feed, forward	29.0	28.8
aft	31.3	31.8
forced-lubrication, starboard	28.8	47.8
port	22.4	7.4
fuel-oil, forward	49.4	50.9
aft	61.5	67.8
Forced-draft blowers (average of four)	1,350.0	1,459.0
Speed of ship, knots	29.601	*30.47

*As determined by extending standardization curves.

	Roe.	*Terry.*
Slip of propeller, in per cent. of its own speed, based on mean pitch :		
Shaft No. 1	20.75	23.78
No. 2	21.47	21.13
No. 3	23.58	24.89
Mean	21.93	23.26
Shaft horsepower :		
Shaft No. 1, starboard	3,596.8	4,407.0
No. 2, center	4,536.37	4,807.0
No. 3, port	3,927.4	4,136.0
(*a*) Total	12,060.6	13,350.0
(*b*) From standardization curve at revolutions corresponding to trial speed	11,789.5	12,700.0
Indicated horsepower :		
Air, circulating and feed pumps	209.4	236.8
All other auxiliaries in operation	298.4	362.8
(*d*) Collective horsepower, main engines and auxiliaries, S.H.P. and I.H.P.	12,297.3	13,949.6
Temperatures, degrees Fahrenheit :		
Injection	76.8	70.0
Discharge	107.0	100.0
Air pumps, discharge	107.0	100.0
Feed	111.0	189.0
Outside air	83.0	74.0
Engine-room working level	132.7	100.0
Fireroom working level, forward	120.0	103.0
aft	112.0	109.8
Smokepipe gases	1,110.0	982.0
Fuel oil :		
Kind	Paraffin base.	
Specific gravity	0.8962	0.902
B.T.U., per pound	19,388.0	19,430.0
(*m*) Pounds per hour	14,937.0	17,712.4
Knots per ton (2,240)	4.438	3.853
Water :		
(*g*) Pounds per hour, all machinery	199,512.5	206,854.5
per S.H.P. main engines	16.923	15.495
Pounds per hour per H. P., all machinery	16.225	14.83

DEDUCED DATA.

(*q*) Square feet of heating surface per S.H.P.	1.527	1.348
(*r*) Pounds of oil per S.H.P. per hour	1.267	1.327
(*s*) H.P. per hour	1.215	1.27
(*u*) Cooling surface (main condensers), square feet per S.H.P.	0.8143	0.719
(*y*) Heating surface used during trial	18,000.0	18,000.0
(*z*) Pounds of water evaporated per pound of oil $\frac{(g)}{(m)}$	13.358	11.678

TWELVE-HOUR TRIALS AT TWENTY-FIVE KNOTS.

ROE.

The twelve-hour twenty-five knot endurance and fuel and water-consumption trial was begun about forty-two miles to the eastward of Cape Henry at 9:30 A. M., July 15, 1910, and was ended twelve hours later at about the same place. The weather was clear and pleasant, with gentle breeze from the S.W. The sea was smooth, with a small swell.

The average displacement for the trial was 711 tons.

TERRY.

The twelve-hour twenty-five knot fuel and water-consumption trial was begun off the Delaware Capes, Five Fathom Bank Light Ship bearing S.W. by W. about six miles distant, at 7:30 A. M., September 22, 1910. The trial was ended at 7:30 P. M., with the ship in nearly the same position. The weather was clear and pleasant, with moderate breeze and light sea from the N.E. Wind and sea moderated as the day progressed. The average displacement of the trial was 724 tons.

PERFORMANCE, TWELVE-HOUR TWENTY-FIVE KNOT TRIALS.

Average steam pressures :	*Roe.*	*Terry.*
At boiler, gage, pounds	256.3	243.0
Engine room, main steam, gage, pounds	254.9	243.0
auxiliary steam, gage, pounds	236.1	253.0
Auxiliary exhaust, gage, pounds	14.4	18.7
On glands, gage, pounds	2.86	3.73
Main H.P. turbine, absolute, pounds	115.7	113.8
H.P. cruising turbine, absolute, pounds	2.53	6.62
I.P. cruising turbine, absolute, pounds	260.8	244.0
L.P. turbine, starboard, absolute, pounds	27.7	28.8
port, absolute pounds	31.3	29.8
Pressure in oil system	...	5.0
Vacuum in condenser, starboard, inches of mercury	29.19	28.0
Vacuum in condenser, port, inches of mercury	29.51	28.0
Air pressure in fireroom, forward, inches of water	1.6	1.5
aft, inches of water	1.5	1.3
Barometer	30.12	...

	Roe.	*Terry.*
Revolutions, or double strokes, per minute:		
Shaft No. 1, starboard	696.707	693.46
No. 2, center	608.822	597.49
No. 3, port	634.63	625.02
Average	646.72	638.65
Pumps, main-air, starboard	24.7	26.7
port	32.4	26.8
circulating, starboard	284.0	283.3
port	281.0	299.0
feed, forward	21.7	20.8
aft	16.5	21.1
forced-lubrication, starboard	21.7	21.4
port	26.0	21.4
fuel-oil, forward	31.4	44.0
aft	44.0	51.4
Forced-draft blowers (average of four on *Roe*; average of two on *Terry*)	880.8	1,293.0
Speed of ship, knots	25.46	25.21
Slip of propeller, in per cent. of its own speed, based on mean pitch:		
Shaft No. 1	23.38	23.7
No. 2	12.32	11.5
No. 3	15.89	15.4
Mean	17.2	16.87
Shaft horsepower:		
Shaft No. 1, starboard	3,183.5	3,173.0
No. 2, center	1,938.9	1,887.0
No. 3, port	1,969.0	1,898.0
(*a*) Total	7,091.6	6,958.0
(*b*) From standardization curve at revolutions corresponding to trial speed	7,122.0	6,550.0
Indicated horsepower:		
Air, circulating and feed pumps	129.1	144.0
All other auxiliaries in operation	126.2	147.9
Total auxiliaries in operation	255.3	291.9
(*d*) Collective horsepower, main engines and auxiliaries, S.H.P. and I.H.P.	7,377.3	7,249.9
Temperatures, degrees Fahrenheit:		
Injection	76.1	68.0
Discharge	98.8	89.3
Air pumps, discharge	101.5	90.0
Feed	104.5	165.0
Outside air	81.8	72.0
Engine room, working level	139.3	97.3
Fireroom, working level, forward	126.0	103.0
aft	118.0	112.0
Smokepipe gases	912.0	671.0

	Roe.	*Terry.*
Fuel oil :		
Kind	Paraffin base.	
Specific gravity	0.8962	0.902
B.T.U., per pound	19,388.0	19,842.0
(m) Pounds per hour	8,931.0	8,864.0
Knots per ton (2,240)	6.386	6.37
Water :		
(g) Pounds per hour, all machinery	116,203.0	115,118.6
per S.H.P. main engines	16.316	16.545
Pounds per hour per H.P., all machinery	15.753	15.89

DEDUCED DATA.

(q) Square feet of heating surface per S.H.P.	2.527	2.59
(r) Pounds of oil per S.H.P. per hour	1.254	1.274
(s) H.P. per hour	1.210	1.221
(u) Cooling surface (main condensers) square feet per S.H.P.	1.35	1.38
(y) Heating surface used during trial	18,000.0	18,000.0
(z) Pounds of water evaporated per pound of oil $\frac{(g)}{(m)}$	13.01	12.99

TWELVE-HOUR TRIALS AT SIXTEEN KNOTS.

ROE.

The twelve-hour endurance and oil-consumption trial at sixteen knots was begun off the Delaware Breakwater at 7:20 A. M., July 14, 1910, and ended near Cape Henry at 7:20 P. M. The weather was overcast and rainy during a portion of the trial, with a light swell. At the end of the trial the sea was smooth.

The average displacement for the trial was 698 tons.

TERRY.

The twelve-hour sixteen-knot fuel and water-consumption trial was begun off Delaware Breakwater at 6:15 A. M., September 23, 1910, and ended about two miles off Cape Henry at 6:15 P. M. The weather was generally fair and pleasant, overcast from 2:00 to 4:00 P. M. There was a gentle to moderate breeze from E.S.E., and smooth sea with light chop.

The average displacement for the trial was 723 tons.

PERFORMANCE OF TWELVE-HOUR SIXTEEN-KNOT TRIALS.

	Roe.	*Terry.*
Average steam pressures :		
At boiler, gage, pounds	241.0	233.0
Engine room, main steam, gage, pounds	232.6	238.0
auxiliary steam, gage pounds	240.6	244.0
Auxiliary exhaust, gage, pounds	10.6	16.0
On glands, gage, pounds	2.3	2.3
Main H.P. turbine, absolute, pounds	27.2	27.0
H.P. cruising turbine, absolute, pounds	211.0	166.0
I.P. cruising turbine, absolute, pounds	62.0	60.0
L.P. turbine, starboard, absolute, pounds	10.0	8.5
L.P. turbine, port, absolute, pounds	10.7	8.5
Pressure in oil system, pounds	...	5.0
Vacuum in condenser, starboard, inches of mercury.	29.3	28.5
port, inches of mercury	29.3	28.5
Air pressure in fireroom, in inches of water	1.1	1.62
Barometer	30.2	...
Revolutions, or double strokes, per minute :		
Shaft No. 1, starboard	391.23	398.876
2, center	346.02	352.835
3, port	403.29	400.18
Average	380.41	383.964
Pump, main air, starboard	39.0	39.0
port	Not in use.	Not in use.
main circulating, starboard	208.0	266.0
port	200.0	273.0
main feed, forward	16.8	Not in use.
after, auxiliary feed	Not in use.	16.0
forced-lubrication, starboard	24.6	Not in use.
port	28.0	35.0
fuel-oil, forward	Not in use.	Not in use.
aft	43.5	60.0
Forced-draft blower (one)	1,381.0	1,362.0
Speed of ship, knots	16.16	16.07
Slip of propeller, in per cent. of its own speed, based on mean pitch :		
Shaft No. 1	13.40	15.5
No. 2	2.08	4.5
No. 3	15.99	15.8
Mean	10.49	11.93
Shaft horsepower :		
Shaft No. 1, starboard	498.43	537.0
No. 2, center	349.93	332.0
No. 3, port	549.69	537.0
(*a*) Total	1,398.05	1,388.0
(*b*) From standardization curve at revolutions corresponding to trial speed	1,490.5	1,200.0

	Roe.	*Terry.*
Indicated horsepower:		
Air, circulating and feed pumps	54.2	79.6
All other auxiliaries in operation	86.0	89.0
Total auxiliaries in operation	140.2	168.6
(*d*) Collective horsepower, main engines and auxiliaries, S.H.P. and I.H.P	1,630.7	1,556.6
Temperatures, degrees Fahrenheit:		
Injection	73.6	69.3
Discharge	85.7	77.5
Air pumps, discharge	84.6	79.0
Feed	93.6	94.0
Outside air	77.6	69.0
Engine room, working level	127.0	98.0
Fireroom, working level, forward	Not in use.	Not in use.
aft	124.0	108.0
Smokepipe gases	801.0	742.0
Fuel oil:		
Kind	Paraffin base.	
Specific gravity	0.8962	0.902
Flash point	370.0	335.0
B.T.U. per pound	19,388.0	19,842.0
(*m*) Pounds per hour	3,182.0	2,662.3
Knots per ton (2,240)	11.377	13.52
Water:		
(*g*) Pounds per hour, all machinery	38,175.0	31,807.8
per S.H.P. main engine	25.612	22.913
Pounds per hour per H.P., all machinery	23.41	20.42

DEDUCED DATA.

(*q*) Square feet of heating surface per S.H.P.	6.039	3.24
(*r*) Pounds of oil per S.H.P. per hour	2.135	1.92
(*s*) H.P. per hour	1.951	1.71
(*u*) Cooling surface (main condensers), square feet per S.H.P.	6.44	6.92
(*y*) Heating surface used during trial	9,000.0	4,500.0
(*z*) Pounds of water evaporated per pound of oil $\frac{(g)}{(m)}$	12.0	11.95

WEIGHTS OF PROPELLING MACHINERY—"ROE."

Items.	Weight in pounds.
Main turbines,	101,200
Shafting,	21,960
Bearings,	3,940
Main condensers,	35,310
Main air and circulating pumps,	19,560

Items.	Weight in pounds.
Propellers,	4,280
Boilers,	153,580
Boiler fittings,	33,090
Smokepipes and uptakes,	18,730
Steam and exhaust pipes and valves, Suction and discharge pipes and valves,	69,890
Lagging and clothing,	9,700
Flooring and gratings,	8,800
Auxiliaries,	39,220
Fittings and gear,	21,260
Stores, tools and spares,	7,260
Miscellaneous machinery,	22,490
Steam and exhaust to auxiliaries,	1,240
Total water,	51,460

GENERAL NOTES ON THE TRIALS.

After the completion of the standardization trials of the *Roe,* July 13, the anchor test was made. The total time required to heave in 96 fathoms of cable was 15 minutes and 53 seconds, which gave a rate of heaving in of 6.04 fathoms per minute. The windlass engine worked satisfactorily, and after the test the windlass thrust, worm and worm wheel were found to be in fair condition.

The anchor test of the *Terry* was made September 20, 1910, while waiting for the visibility of the trial-course ranges. The heaving in of the cable was at the rate of 6.16 fathoms per minute. The test of the *Terry's* anchor gear was satisfactory, excepting a slight cutting of the teeth of the composition worm.

The twelve-hour sixteen-knot trial of the *Roe* was run with five turbines in operation, one main air pump, one fireroom with one blower only, and, for nine hours of the trial, two boilers in operation.

The generator exhaust went to the auxiliary exhaust line. The auxiliary exhaust, at an average pressure of 10.5 pounds,

was all put into the second stage of the main H.P. turbine. The feed heater was not in use.

During the last part of the trial, from 4:30 P. M. until 7:20 P. M., only one boiler was in use. Immediately after the completion of the twelve-hour trial, at 7:20, the second boiler was cut in again in order to increase the speed for running in from the trial course to Newport News, as it was desired to run in at about 25 knots speed. The cutting out and restarting again of the boiler during the run shows the flexibility of this oil-burning system. It was also noticed that with one boiler in operation the fuel-oil consumption was somewhat lower, due to the better regulation which was possible when running one boiler at a higher power than when operating two boilers at such small powers.

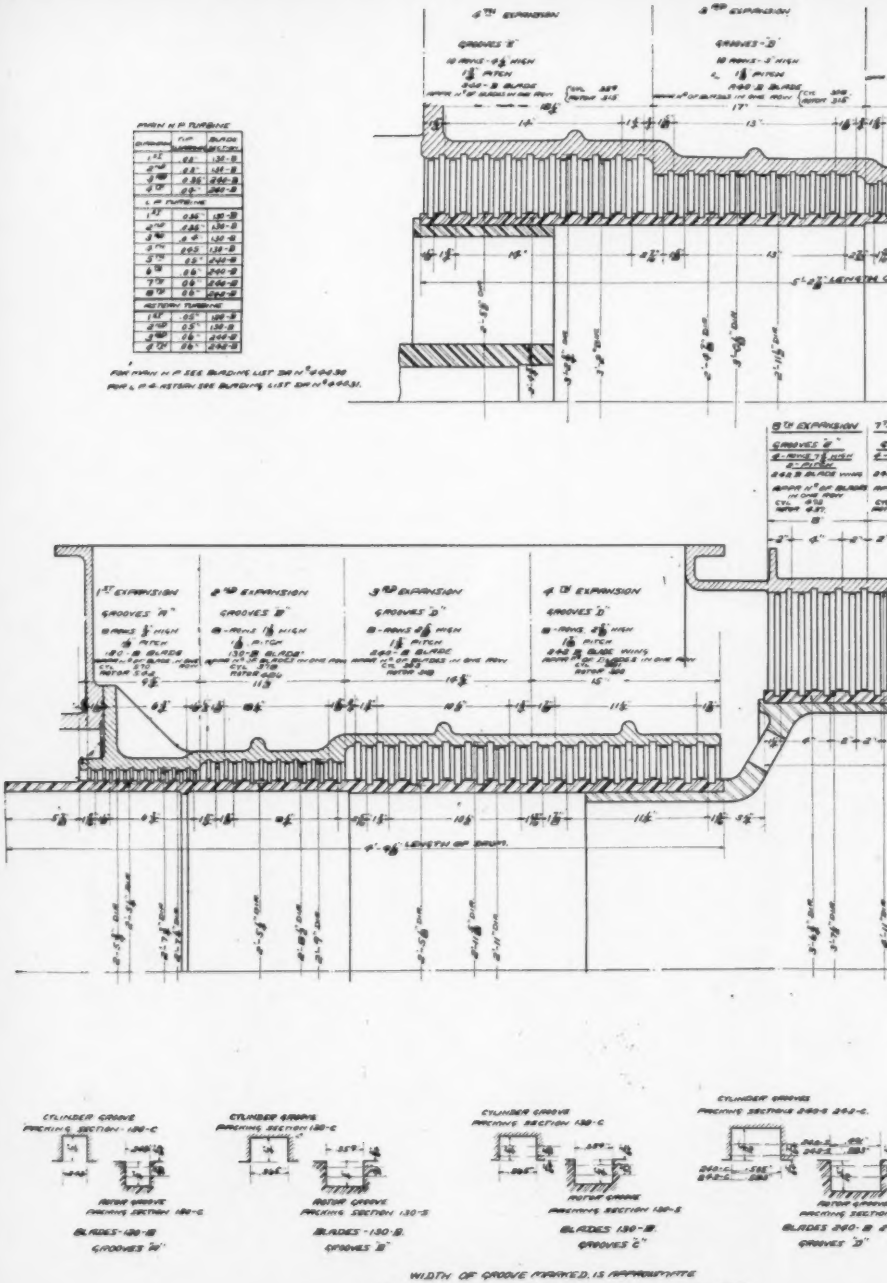
During this trial it was found that slight modifications could be made in the details for regulating the air doors in the fronts of the boiler casing supplying air to the oil burners, which would give better regulation. This modification was carried out later on the *Roe*, and was incorporated in the work before the trials of the *Terry*.

The twelve-hour sixteen-knot trial of the *Terry* was run with five turbines in operation, one air pump, one blower and one boiler.

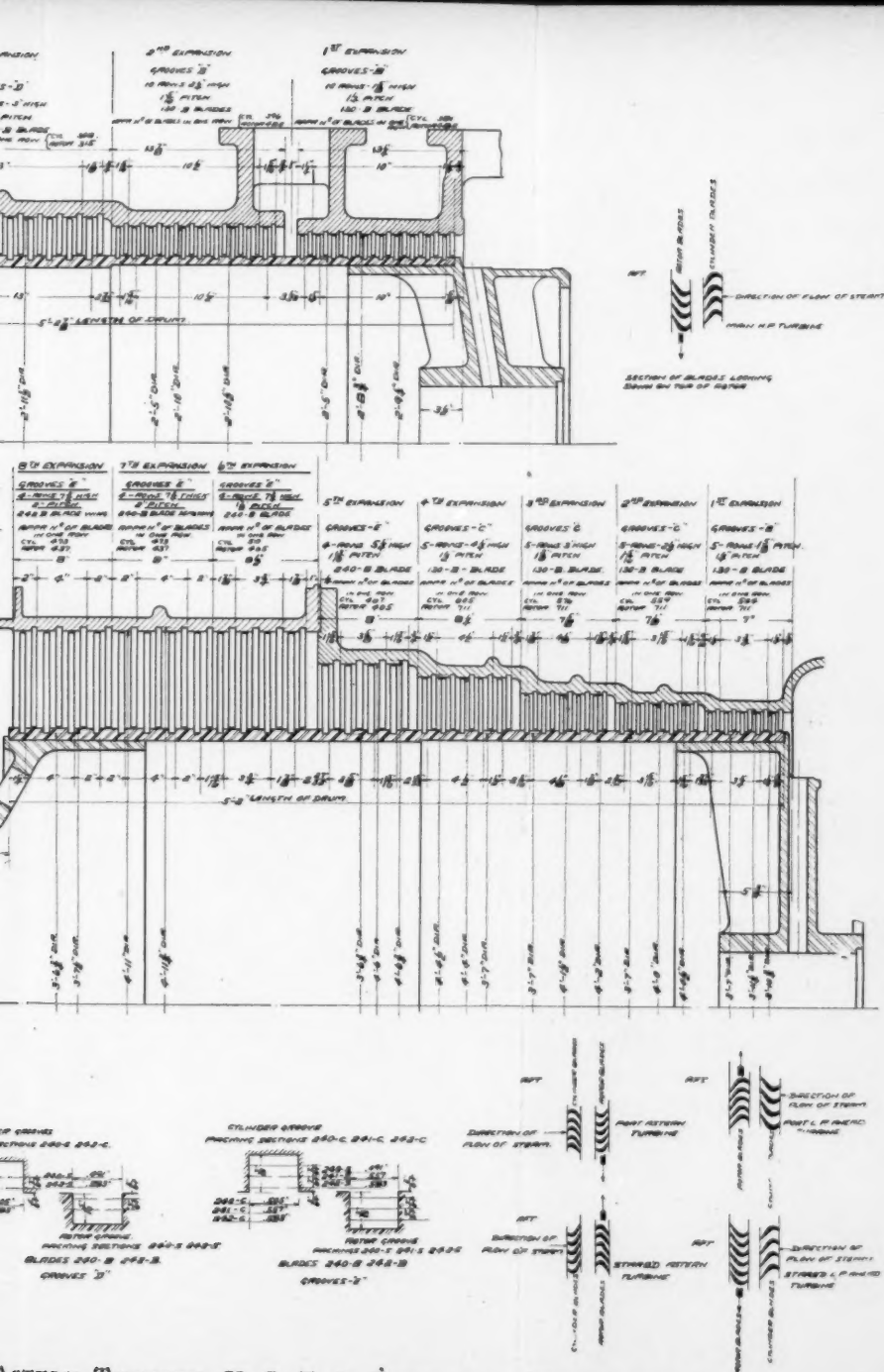
The twelve-hour twenty-five-knot trial of the *Roe* was run with four turbines in operation; all the boilers were in use as well as all the blowers and both the air pumps.

The generator exhaust was run to the auxiliary exhaust which was opened to both L.P. turbines and cracked open on the feed-water heater. The pressure in the auxiliary exhaust line in the engine room was about 15 pounds. The feed-water heater drain was led to the port condenser. After the trial was finished, at 9:30 P. M., outside the Capes at Chesapeake Bay, the vessel proceeded to the shipyard at Newport News at a greater speed than was made on the trial. The *Roe* was tied up at her berth at the shipyard at 1 P. M., July 16, 1910.

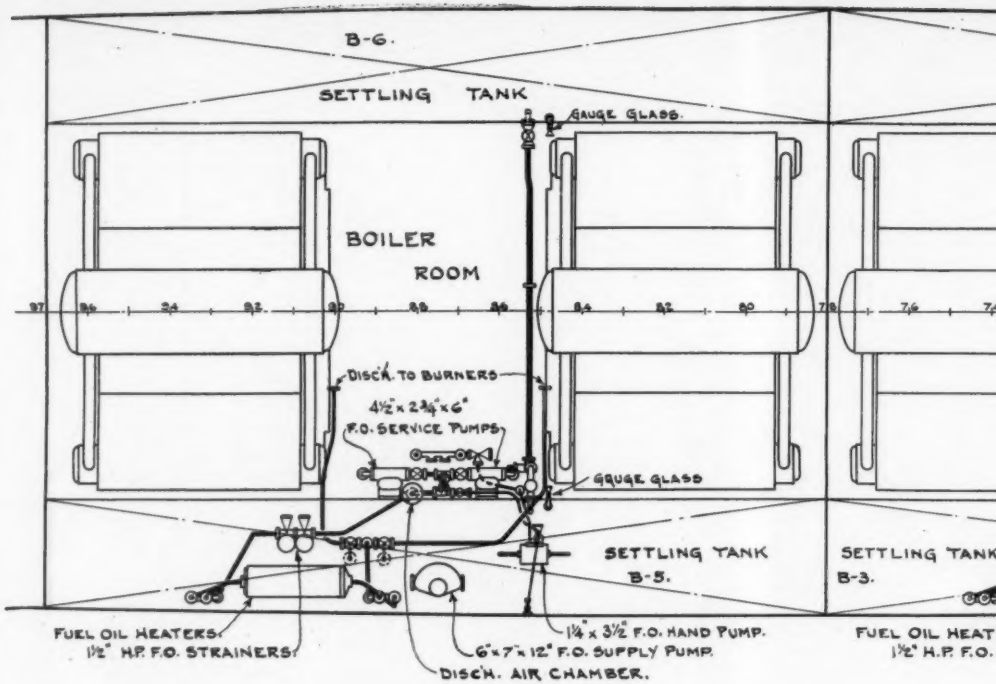
The twelve-hour twenty-five-knot trial of the *Terry* was

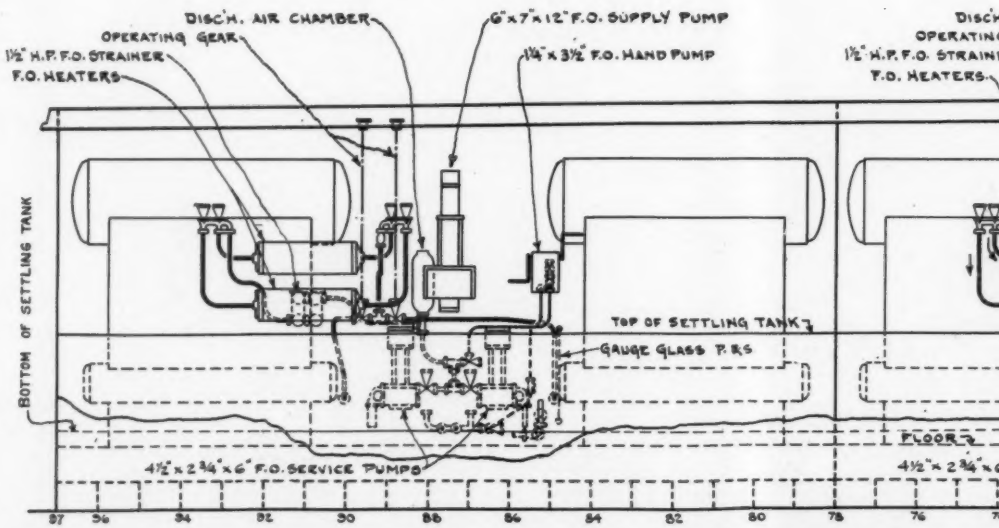


ARRANGEMENT OF BLADING FOR MAIN H.P., L.P. AND ASTERN TU



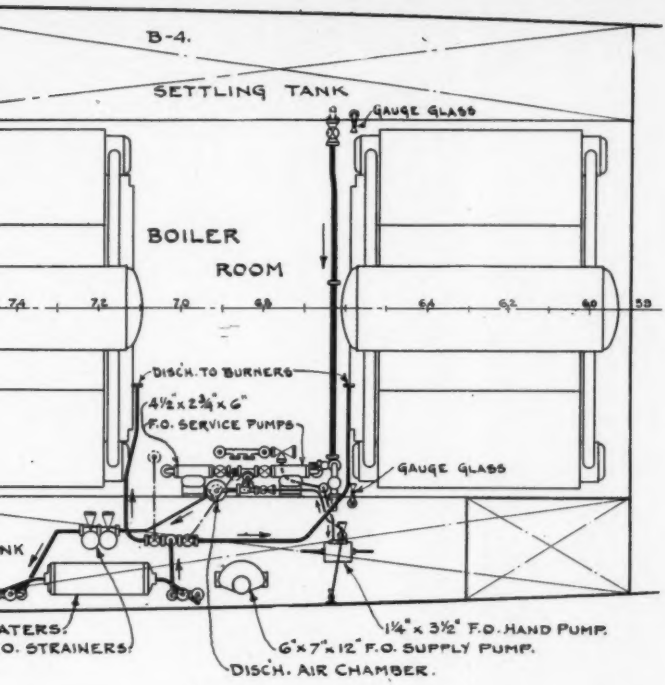
ASTERN TURBINES, U. S. T. B. DESTROYERS "ROE" AND "TERRY."

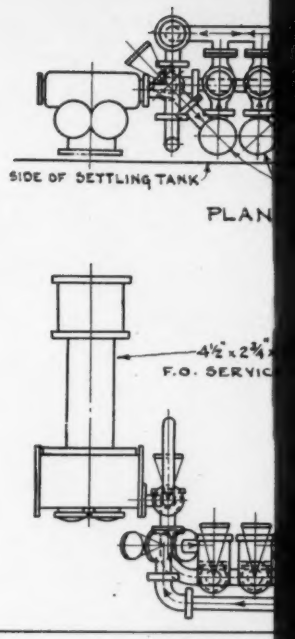


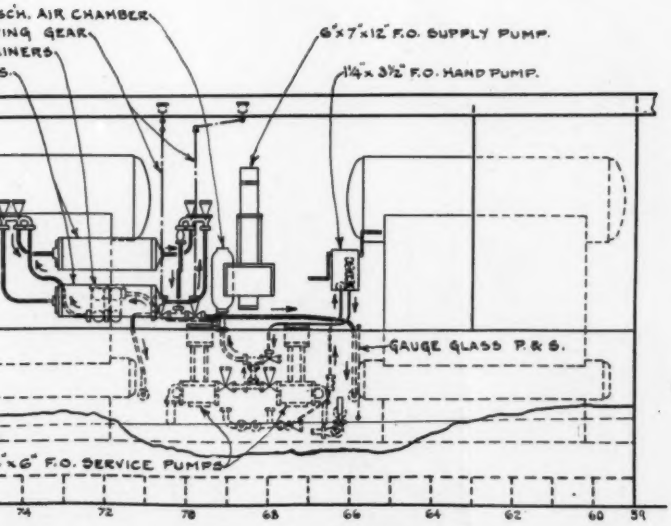


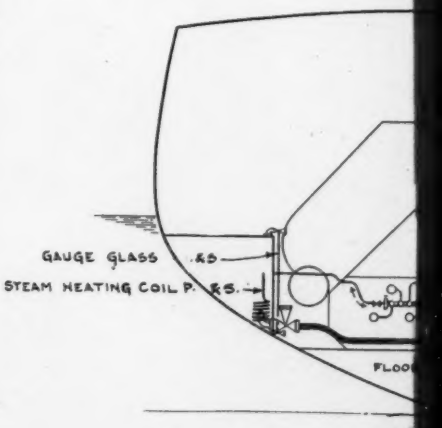
ELEVATION STAR'B'D. SIDE LOOKING INBOARD TO PORT

ARRANGEMENT OF FUEL-OIL



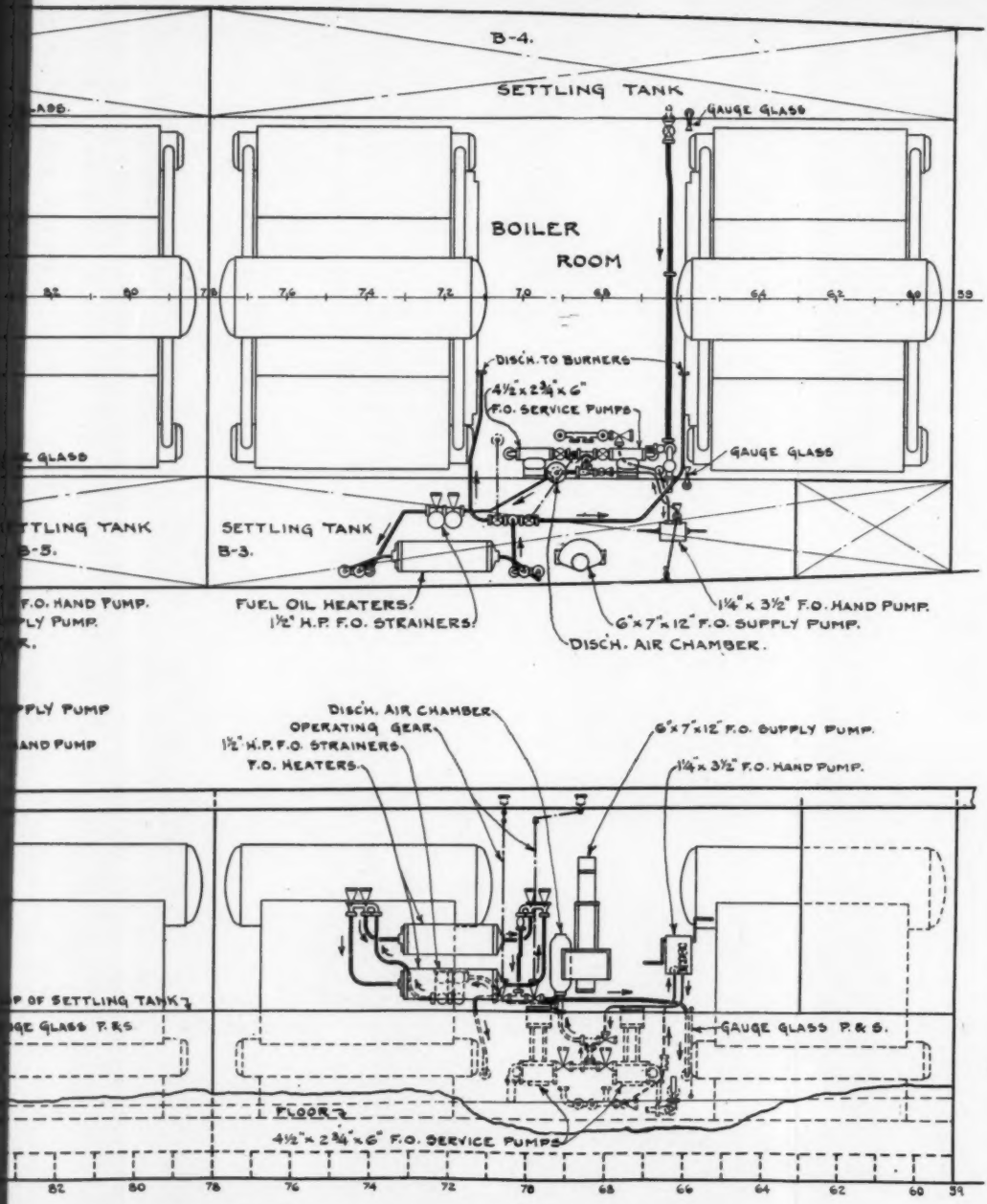






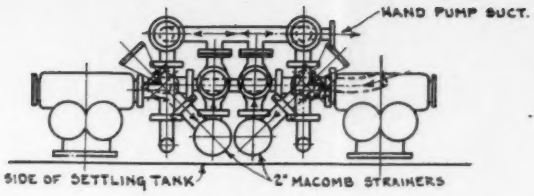
T SHELL PLATING REMOVED.

OIL DISCHARGE PIPING, U. S. T. B. DESTROYERS "ROE" AND "TERRY."

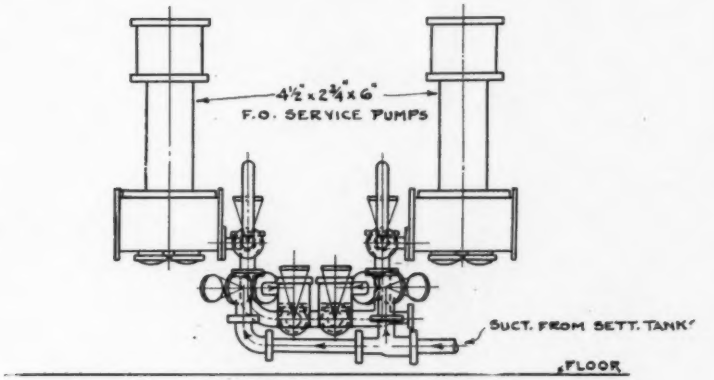


LOOKING INBOARD TO PORT SHELL PLATING REMOVED.

ARRANGEMENT OF FUEL-OIL DISCHARGE PIPING, U. S. T. B. DESTROYERS "ROE" AN

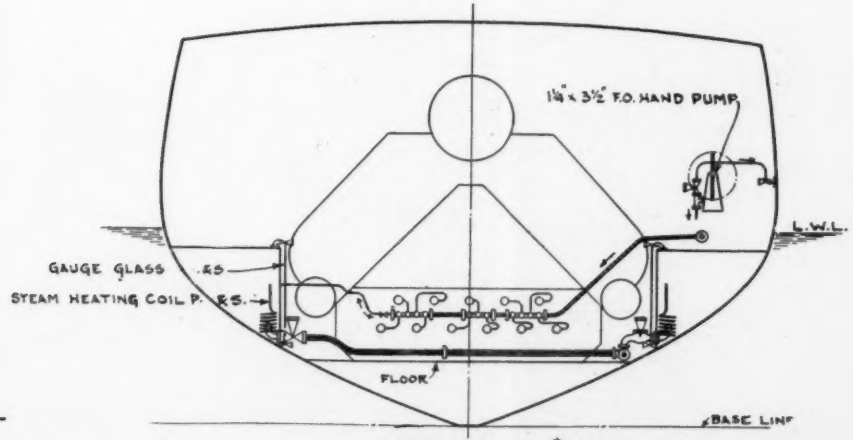


PLAN



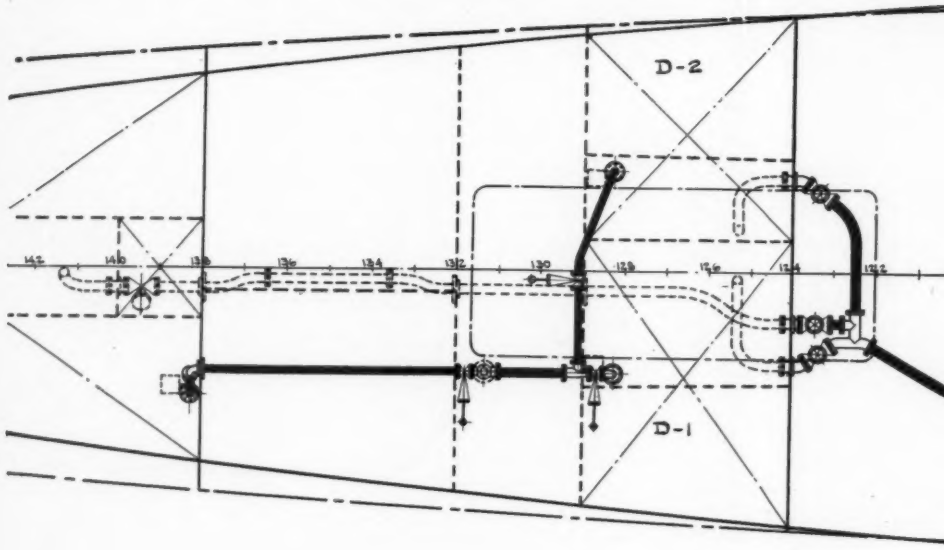
ELEVATION.

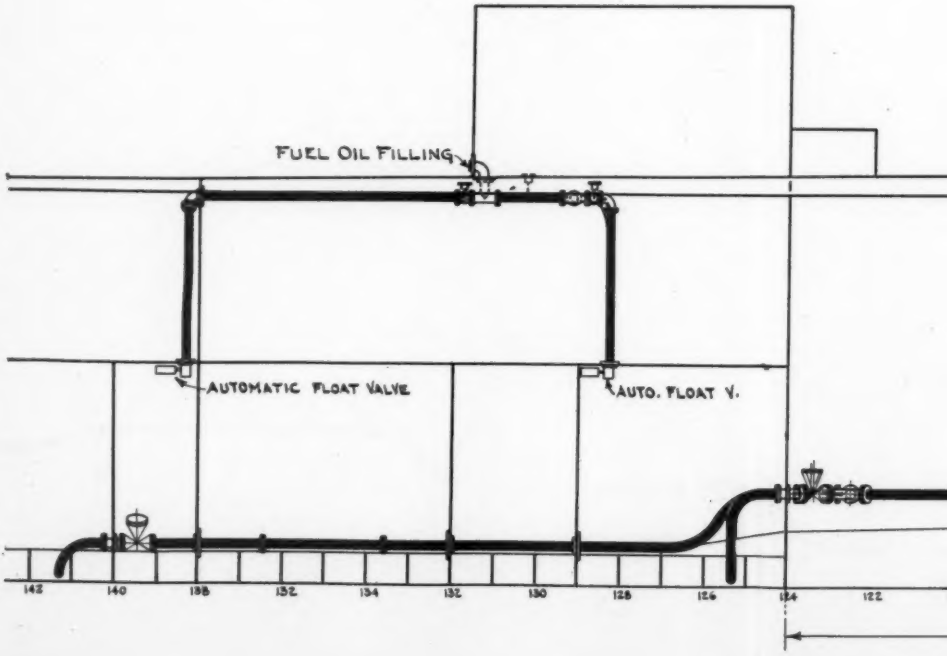
SKETCH SHOWING F.O. SUCT. CONN'S. AT PUMPS.

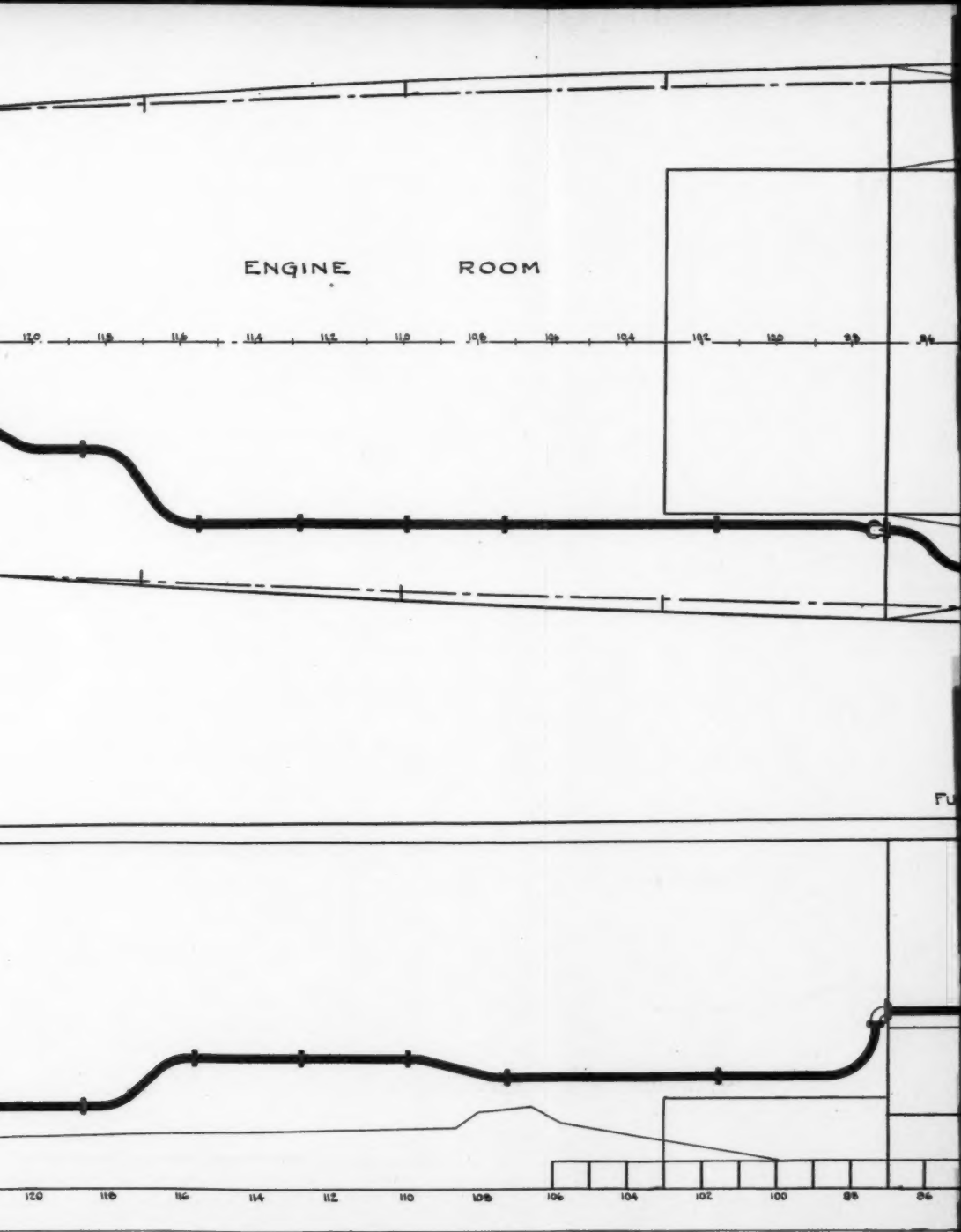


SECTION AT FR.* 67
LOOKING FORWARD.

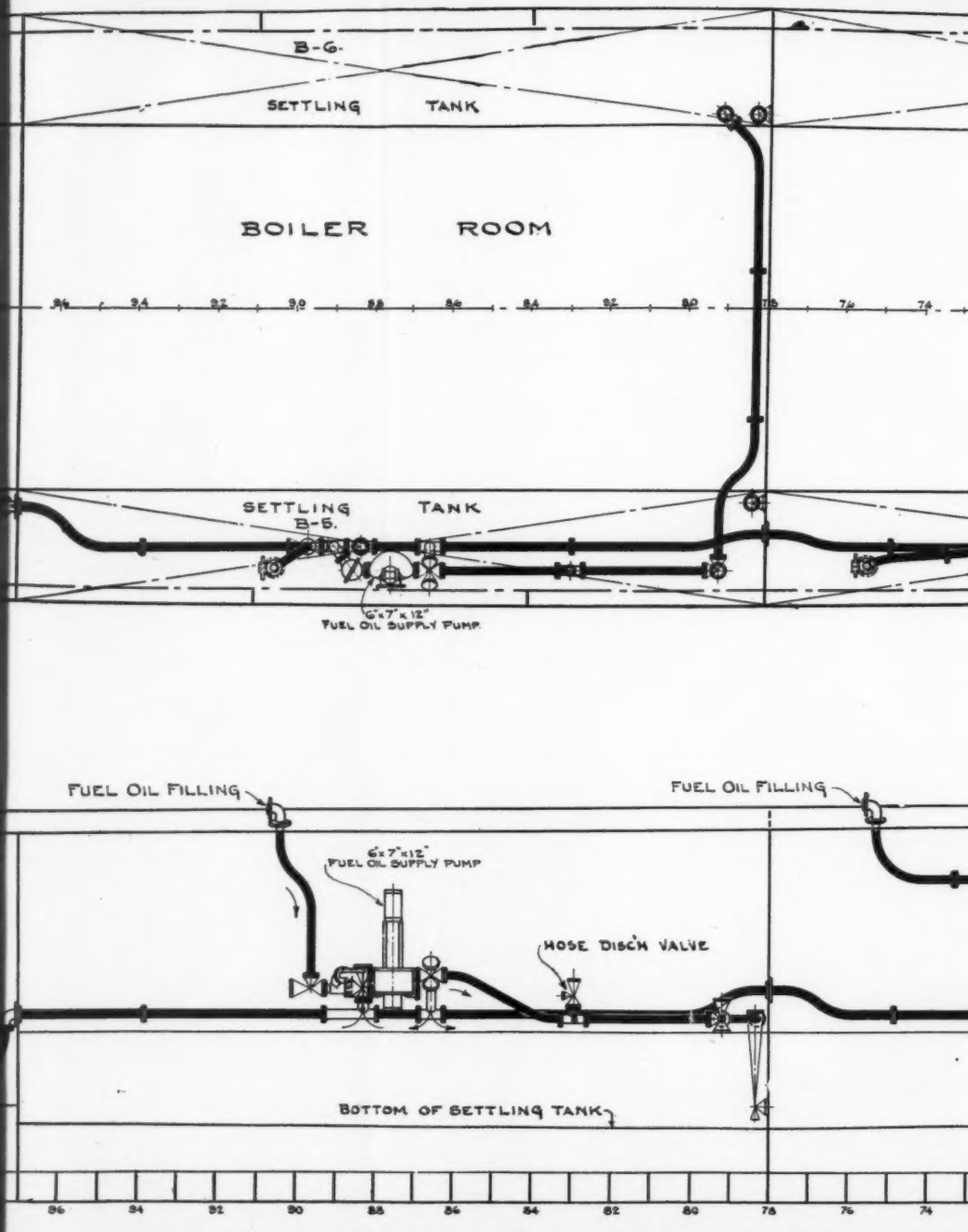
AND "TERRY."


D-2
D-1


FUEL OIL FILLING
AUTOMATIC FLOAT VALVE
AUTO. FLOAT V.
142
140
138
132
134
132
130
128
126
124
122

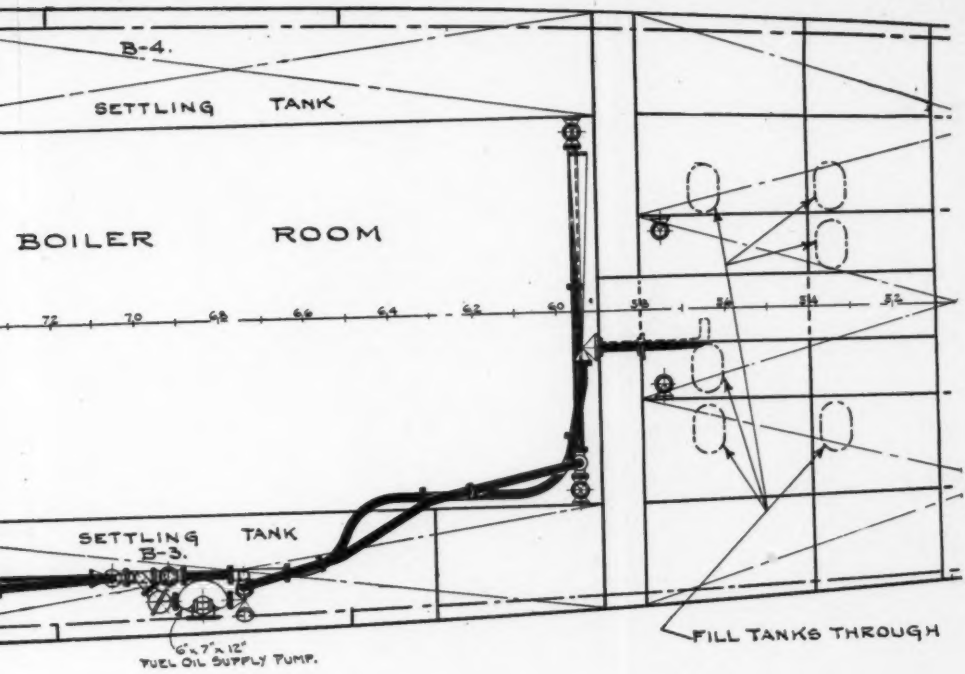


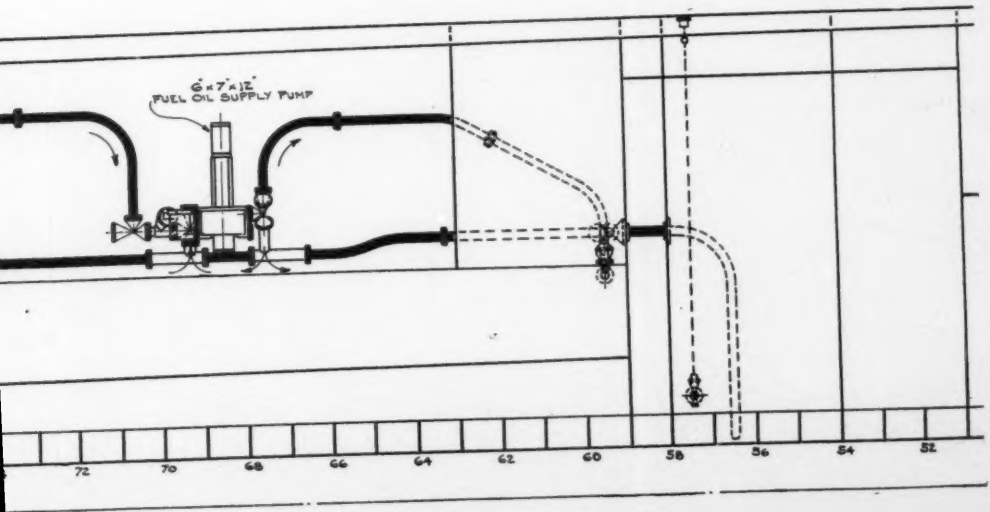
ARRANGEMENT OF FUEL-OIL FILLING AND SUCTION PIPES,



ELEVATION STARBOARD SIDE LOOKING INBOARD TO PORT.

PIPES, U. S. T. B. DESTROYERS "ROE" AND "TERRY."


B-4.
SETTLING TANK
BOILER ROOM
72
70
68
66
64
62
60
58
56
54
52
SETTLING TANK
B-3.
6" x 7" x 12"
FUEL OIL SUPPLY PUMP.
FILL TANKS THROUGH


6" x 7" x 12"
FUEL OIL SUPPLY PUMP
72
70
68
66
64
62
60
58
56
54
52

run with four turbines in use and during most of the trial the by-pass of the I.P. cruising turbine was wide open. The four boilers were in use, but only one blower in each fireroom. The auxiliary exhaust was run on the port low-pressure turbine and the excess, after regulating to about 20 pounds, was run to the feed heater. The generator exhaust was led to the auxiliary exhaust.

When running from Newport News on July 16, 1910, outside the Chesapeake Capes to the trial course for the four-hour full-speed trial of the *Roe,* the contractors ran a twenty-five-knot speed trial of 1½ hours' duration, from 9:05 A. M. to 10:35 A. M., without the cruising turbines, in order to determine the water consumption for this speed under these conditions. The data of this trial are as follows:

Mean revolutions	640
Average boiler pressure, pounds	251
pressure at engine, pounds	245
fireroom air pressure, inches	1.7
vacuum, inches	28.65
Total main engine S.H.P. (as measured)	7,069
from standardization curve	6,830
Average water consumption, all purposes, pounds per hour	136,979
Water per S.H.P. per hour, main propelling machinery (as measured on trial)	19.38
Speed, in knots	25.23

A comparison of the water rates of this trial and the twelve-hour trial at about the same speed shows the better economy of the four-turbine combination.

The four-hour full-speed trial of the *Roe* was run with three turbines in operation and all boilers and blowers were in use. Soon after starting on the run the starboard low-pressure thrust bearing began to warm up excessively; water was kept on it, but it was impossible to reduce the temperature

permanently. The trial was finished with a fair margin over the required speed of 29½ knots.

The generator exhaust was taken to the auxiliary exhaust line and the auxiliary exhaust was led to both the main condensers. The feed-water heater was in use and its drain led to the port condenser.

After the completion of the four-hour full-speed trial of the *Roe,* owing to the condition of the starboard low-pressure thrust bearing, the contractors requested that the reversing trials and the helm test, which it was intended to hold immediately after the completion of the four-hour trial, be postponed until some future date. The Trial Board assented to this request.

The backing and steering trials of the *Roe* were held in the Chesapeake Bay on August 31st, 1910, and were generally satisfactory. The helm being amidships and the turbines making about 810 revolutions per minute ahead, the signal was given full-speed astern, the starboard shaft was stopped in 21½ seconds and the port shaft in 22½ seconds; the ship was dead in the water in 56 seconds and the astern turbines were making full speed in 1 minute 10½ seconds. A few minor additions and modifications to the steering gear were recommended by the Board.

The four-hour full-speed trial of the *Terry* was run September 24, 1910, with the three-turbine combination. The mean revolutions of the shafts per minute on the trial were higher than the mean revolutions of the five high-speed runs of the standardization trial. The mean r.p.m. corresponds to a speed of 30.47 knots from the standardization curves, but the official record of speed of the four-hour trial was taken as the speed made on the five high standardization runs. No undue heating of any of the bearings took place. The auxiliary exhaust was on the feed-water heater and the main condenser during the trial. The dynamo exhaust went directly to the main condenser. All the boilers and blowers were in use.

The steering-gear tests of the *Terry* were made September

20, 1910, while waiting for the weather to clear. The tests of this gear were generally satisfactory. A few minor recommendations were made by the Board. With the vessel making about 29½ knots, with the helm amidships, the engine telegraphs were thrown full-speed astern. The time from the signal to dead in the water was 1 minute 6 seconds. The time the signal to reverse was received in the engine room until the turbines were turning astern was 33 seconds for the starboard shaft and 35 seconds for the port shaft. The engines were running full-speed astern in 1 minute and 37 seconds after the signal to reverse was received.

During the run of the *Terry* from Newport News to the Delaware Breakwater the thrust on the port shaft heated. After anchoring it was lifted and found to be slightly scored. The collars were scraped to a bearing, but during the standardization and steering trials on September 20, 1910, this bearing heated again slightly. During the standardization trials on September 21st the starboard low-pressure thrust heated on the high-speed runs; after the completion of the standardization trials the caps of both the low-pressure thrust bearings were lifted and the bearings scraped, after this these bearings gave no further trouble. It is to be noted that the material of the thrust rings and thrust bearings was of composition H.

During all the trials of both the *Roe* and *Terry* it was the aim of the contractors to run the boilers with a very slight haze of smoke appearing at the stacks. It was quite possible to run the boilers on any of the trials with practically no smoke appearing, but it was not considered best to do so, owing to the liability of heating and discharging an excess of air through the boilers. Unfortunately, no photographs were taken of the vessels during any of the trials; such photographs would have emphasized the freedom from smoke. This absence of smoke, however, is shown by photographs taken during a preliminary test of the boilers conducted April 14, 1910, on the boilers in the forward fireroom of the *Roe* with about 3 inches to 3¼ inches air pressure, corresponding

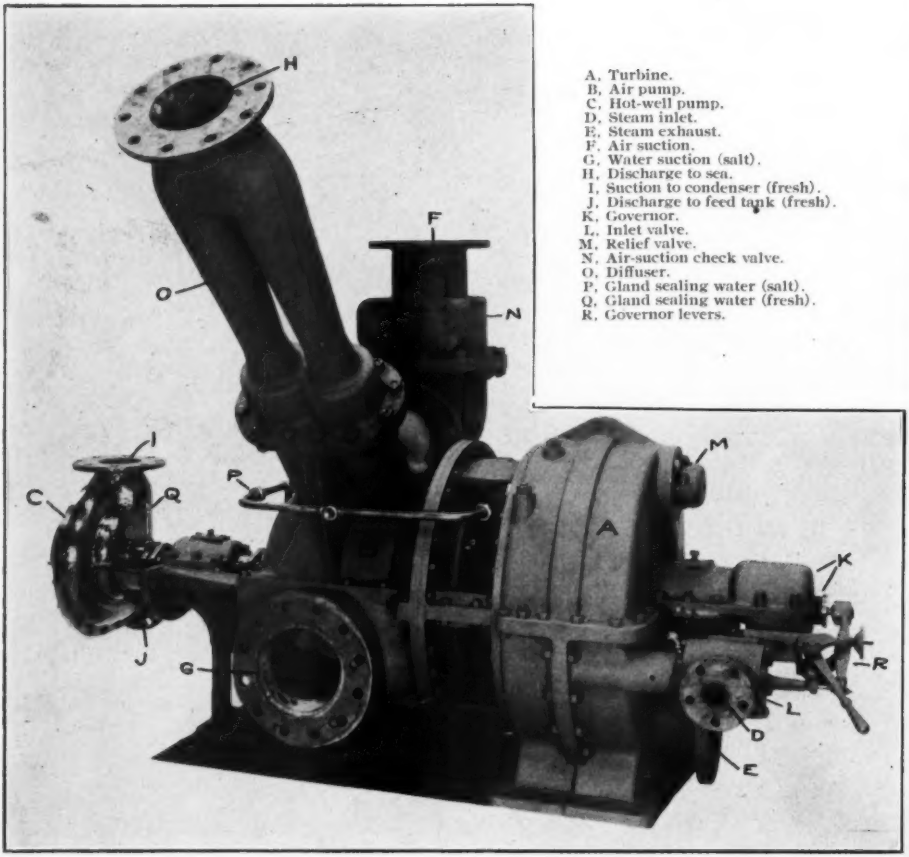
closely to the air pressure maintained during the four-hour full-speed trials.

An examination of the machinery of both vessels after the trials showed that the interior of all boilers, drums, tubes, pipes, etc., were clean and these parts tight.

The fire surfaces of the boilers were remarkably free from soot deposits. The brick work of the furnace sides and bottoms was found in good condition, that on the fronts in fair condition above the air cones; the back walls were warped somewhat from their vertical planes and the cement burned and loosened to some extent. The air cones of the burners were slightly warped and somewhat burned on the *Roe*, but were found in good condition on the *Terry*. All the turbine casings were lifted and the rotors revolved, the blading was in excellent condition. The starboard thrust bearing on the *Roe*, which ran hot during the four-hour full-speed trial, was in such condition as to necessitate the renewal of the composition rings. These rings were originally fitted of composition H, but the replace rings were ordered of composition —copper, 6; tin, 1; lead, ¼, corresponding to copper, 83; tin, 13; lead, 4.

The thrust-shaft collars of this bearing were uninjured and in good condition; the port shaft was in good condition, but the four after rings showed signs of some heating. All of the auxiliary machinery, including blower engines, pumps, generators, condensers, etc., were in excellent condition.

During all the trials the vessels were remarkably free from vibration.



A, Turbine.
B, Air pump.
C, Hot-well pump.
D, Steam inlet.
E, Steam exhaust.
F, Air suction.
G, Water suction (salt).
H, Discharge to sea.
I, Suction to condenser (fresh).
J, Discharge to feed tank (fresh).
K, Governor.
L, Inlet valve.
M, Relief valve.
N, Air-suction check valve.
O, Diffuser.
P, Gland sealing water (salt).
Q, Gland sealing water (fresh).
R, Governor levers.

THE LEBLANC AIR PUMP SET FOR THE U. S. S. "NEPTUNE."

THE MARINE-TYPE LEBLANC AIR PUMP.

By Lieutenant W. W. Smith, U. S. N., Member.

To realize the high economy resulting from a high vacuum, the development of the marine turbine has led to many improvements in the condensing plant. In obtaining the high vacuum required, one of the chief difficulties has been the removal of air from the condenser, and this has been overcome generally by modifying the air pumps used for producing the moderate vacuum of engine practice. Although the changes have greatly improved the vacuum, the air-withdrawing machinery requires more weight, space and frequently more steam than is desirable.

For marine work in general and for naval installations in particular, there is a need of high vacuum efficiency together with economy in steam, lightness and compactness.

The Leblanc air pump fulfills excellently these requirements; and, if, in marine work, it should meet with as much success as it has on land; it is destined to have an important place among marine auxiliaries.

The following notes contain a description of the principle features of the Leblanc air pump and suggestions for its operation.

Although the Leblanc air pump is extensively used on land, it is new for marine work and practical experience is limited to the lessons learned in shore practice; on which, the following notes are principally based. Experience may teach that some of these suggestions are not the best practice and that omissions have been made; but it is not possible to foresee all conditions of operation; and, consequently, the engineer should, as in all cases, follow the best practice as learned from actual experience.

PRINCIPLE OF OPERATION.

Leblanc Air Pump.—As shown in Figs. 1, 3 and 5, air is drawn from the condenser and water from the sea by the air pump which discharges a mixture of the air and water back to the sea. Sea water is the medium used for withdrawing the air. The fresh water condensed from the steam entering the condenser does not enter the air pump, but is withdrawn separately by the hotwell pump.

The air pump consists of a rotor which revolves in an annular space between the water chamber and the casing. The water chamber is supplied through the suction pipe which is connected with the sea. On one side of the water chamber, there is a nozzle which directs the water against the blades of the rotor, which are so designed that the rapid movement of the blades across the stream of water flowing from the nozzle slices off layers from the stream and projects them with considerable force into the nozzle of the discharge pipe. These layers form a continuous series of water pistons which enter the discharge nozzle with sufficient velocity to pass out through it and the discharge pipe to the sea, overcoming the resistance due to the pressure of the atmosphere and of the water.

In slicing off the layers from the continuous stream flowing from the nozzle, the blades produce vacuous spaces, or pockets, between the layers, in which, the air contained in the nozzle is imprisoned as the layers are projected out from the blades into the nozzle. The layers, or water pistons, have sufficient tenacity to carry with them the air imprisoned between, and thus the alternate layers of water and air enter the nozzle and pass out through the discharge pipe. In passing out through the nozzle, the layers of air are compressed by the succeeding layers of water, and finally the layer formation is destroyed, causing the water and air to mingle; the continuously-succeeding layers and the continuous flow through the nozzle causing the mixture of air and water to continue its passage through the discharge pipe.

The vacuous spaces continuously formed by the projection

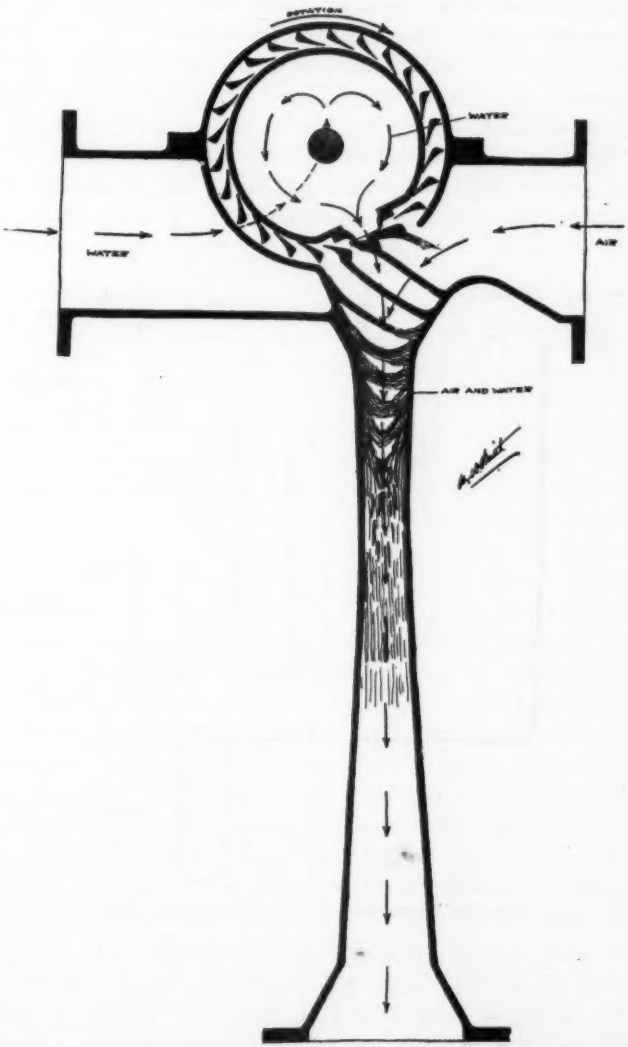


Fig. 1.—Diagram of Leblanc Air Pump, Showing Principle of Operation.

of the layers of water are filled by air which flows in through the suction pipe. Thus the succeeding pockets imprison the air which flows in behind the preceding water pistons, and the succession of layers is so rapid that the flow of air is practically continuous.

The vapor, which is always present in the air drawn from a condenser; in passing through the nozzle, is rapidly condensed by its contact with the water.

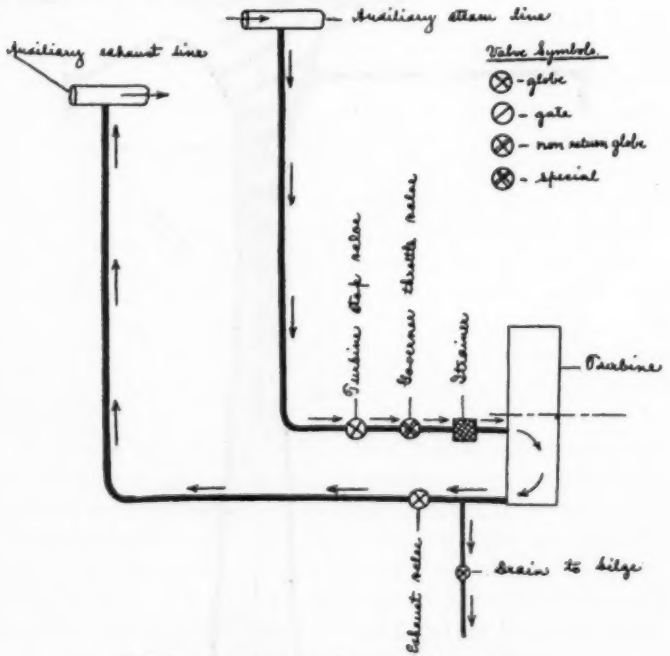


Fig. 2.—Diagram of Turbine Connections.

As the water chamber is in communication with the parts under vacuum, the water is drawn in by suction. Where the pump is located below the water line, as it will be in nearly all cases; the pressure will initially cause the water to flow into the pump.

The action of the water pistons in the Leblanc air pump

constitutes a pump of no clearance, and consequently the vacuum efficiency is almost ideal. In its rotary action, this pump resembles the steam turbine, and it may be truly called a turbine-type air pump.

Hotwell Pump.—The hotwell pump, which is of the ordinary centrifugal type, draws the water condensed from the steam from the condenser and discharges it into the feed tank. When no water is flowing through the pump, the centrifugal action of the impeller will prevent the water in the feed tank from backing up into the condenser.

Turbine.—The turbine is of the impulse type, having one rotor with one row of blades. The steam enters through the governor-controlled throttle valve, where it is automatically throttled as required to maintain a constant speed. It then enters the steam chest; passes into the diverging nozzle, where it is expanded in one drop to the pressure of the exhaust chamber; and is discharged from the nozzle into the rotor buckets at high velocity. In the rotor buckets, the direction of motion is changed and, passing out of the buckets at reduced velocity, the steam enters the reversing chamber, where it is re-directed into the buckets. During the second passage through the buckets, its direction of motion is again changed, its velocity being further reduced; so that, as it passes into the exhaust chamber, the residual velocity is small.

As the turbine is designed for exhausting against a back pressure of ten pounds, it cannot efficiently utilize a high vacuum, and consequently, the gain due to a high vacuum will be small; which makes it desirable to run it under the designed conditions. The best economy will be obtained by exhausting into the auxiliary-exhaust system and utilizing the exhaust for feed-water heating; or by running the exhaust into the L.P. stage of the main turbine.

The governor is of the centrifugal type and operates the throttle valve directly through levers. To allow for variations of steam pressure and for the operation of the governor, the nozzle is designed for a working pressure of about seven-tenths of the main working pressure, so that a constant speed will be maintained under all conditions.

CONSTRUCTION AND INSTALLATION.

Pipe Connections, etc.—The pipe connections, valves, etc., are shown in Figs. 2, 3 and 4. The following particulars should be noted:

Turbine.—The strainer, although shown separately for clearness, is contained in the governor-throttle valve. The drain is led to the bilge, so that the interior can be kept dry when not in use.

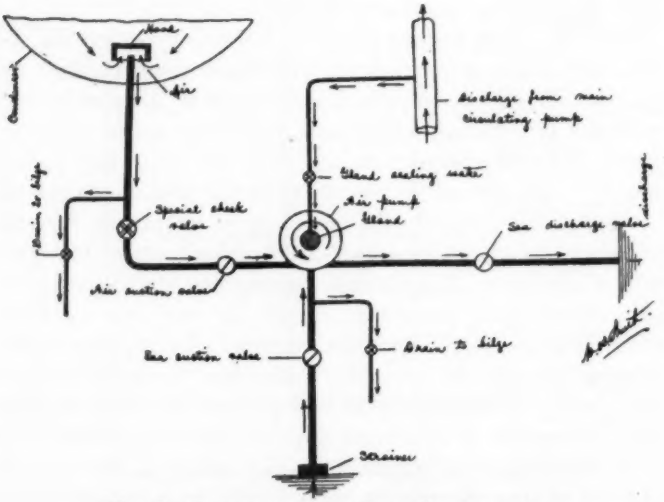


Fig. 3.—DIAGRAM OF AIR-PUMP CONNECTIONS.

Air Pump.—A hood is fitted over the air-suction pipe in the condenser to prevent the loss of fresh water. A special check valve is located in the air-suction pipe to prevent salt water from accidentally backing up into the condenser. The suction and discharge valves are of the gate type to allow free flow. A drain is placed on the air-suction pipe to indicate accidental leakage of salt water, and another is located on the water-suction pipe, so that the pump can be kept dry when not in use. To prevent foreign substances from entering the pump, a strainer is located on the sea suction. (The openings

should not be larger than three-eighths of an inch.) The gland-sealing water is taken from the discharge from the main circulating pump, as shown. As far as possible, the air-suction and the sea-discharge pipes should be free from restrictions, and the latter should be straight; or, if bends are necessary, they should be large.

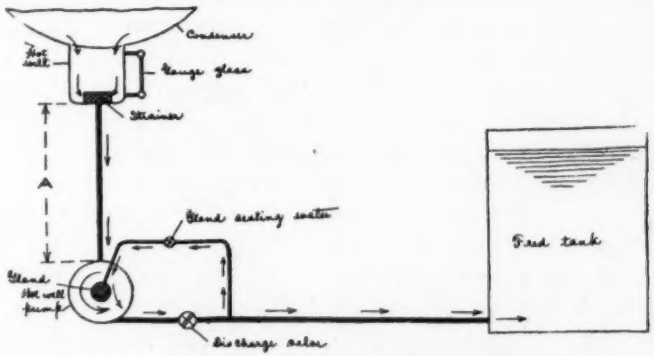


Fig. 4.—Diagram of Hotwell Pump Connections.

Hotwell Pump.—A small hotwell, fitted with a gage glass and suction strainer, is located under the condenser. As there is practically no pressure in the condenser, the water flows to the pump by gravity only; and, as it is necessary to have sufficient pressure to cause the water to flow into the pump, there must be sufficient head. The head (marked A on the diagram) should not be less than 3 feet. The suction pipe should be short and without restrictions. The non-return valve, located in the discharge pipe, prevents the water in the feed tank from accidentally backing up into the condenser when the pump is stopped; and it also serves to cut out the pump when not in use. The gland-sealing water is taken from the discharge pipe, as shown.

The method of installing will vary considerably with the engine-room arrangement, and it should be understood that the diagrams are only intended to cover the general principles.

The suction and discharge pipes of the air pump may be

A, Turbine.
B, Air pump.
C, Hotwell pump.

Turbine.

D, Inlet.
E, Exhaust.
F, Governor.
G, Automatic-safety stop.
H, Bearing.
I, Guard ring.
J, Gland.
K, Rotor.
L, Bucket.
M, Relief valve.
N, Drain.
O, Oil well.

Air pump.

P, Air suction.
Q, Sea suction.
R, Sea discharge.
S, Distance piece.
T, Drain.
U, Gland.
V, Sealing-water connection (sea water).
W, Spacer.
X, Water chamber.
Y, Distributing nozzle.
Z, Rotor.
a, Blade.
b, Discharge nozzle.
c, Starting nozzle.
d, Diffuser.
e, Thrust bearing.
f, Washer.
g, Liners.
h, Thrust collar.
i, Nut.

Hotwell pump.

j, Suction.
k, Discharge.
l, Rotor.
m, Gland.
n, Sealing-water connection (fresh water).
o, Plug.

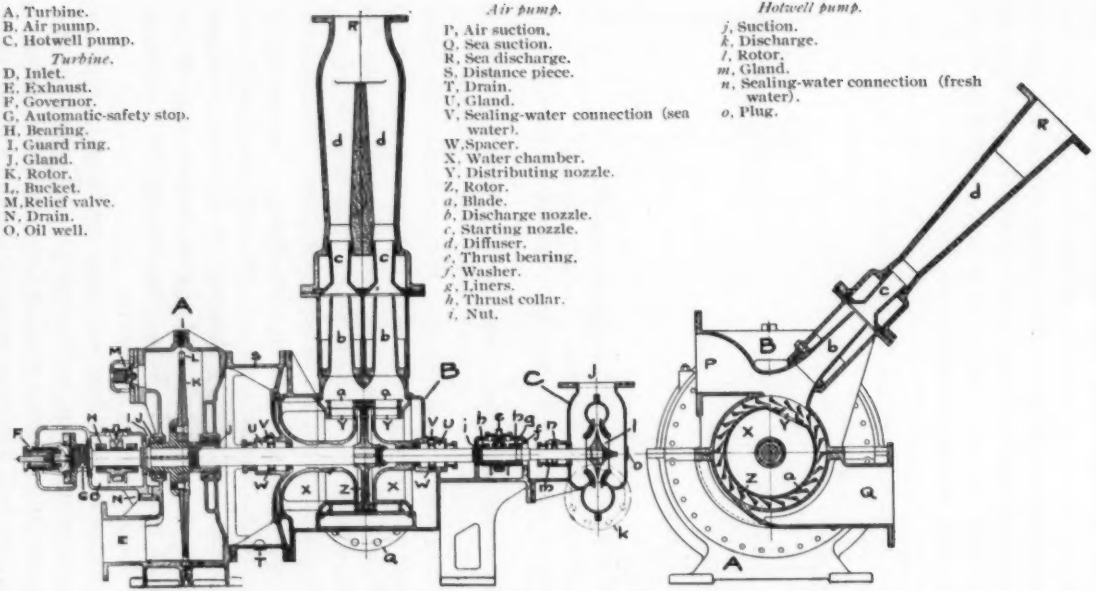


Fig. 5.—Marine-Type Leblanc Air Pump.

connected to convenient sea suction and discharge pipes. The discharge from the air pump may be directed upward or downward at the required angle. The location of the drain on the air pump depends on the design, and should be placed to give the best drainage.

Casings.—The turbine and air-pump casings are divided horizontally, all piping being connected to the lower parts, so that the covers are free for lifting. In case it is necessary to direct the discharge of the air pump upward; the discharge, and also the air-suction pipe, will be connected to the upper part of the casing. In most cases, however, these connections will be made to the lower part. The casing of the hotwell pump is divided vertically and removed from the end. The bearings are carried by the casing, as shown.

No gaskets are used in the joints of the turbine casing; the surfaces being scraped true, coated with boiled linseed oil and fitted together metal to metal. Thick paper is used for making the pump joints; and, in addition to this, the exterior parts are coated heavily with paint to seal them against air leakage.

Glands.—The turbine glands are packed with composition split rings which fit into grooves on the hub of the rotor, as shown. The outer surfaces of the rings bear against a cast-iron sleeve, which is not divided. Steam for sealing the gland against air leakage may be admitted through the steam passage and annular space around the sleeve; but this is not necessary where the turbine is exhausting under a back pressure. A guard ring is fitted on the shaft outside of the end gland to prevent steam or water from blowing into the bearing.

The air-pump glands are packed with soft packing; each gland having two sections of soft packing, separated by a composition spacer. Water for sealing the glands against air leakage is led from the discharge of the main circulating pump, as shown. Each packing ring consists of an inner and an outer ring fitting together conically.

The gland of the hotwell pump is similar to those of the air pump, with the exception of being sealed with fresh water.

Bearings.—The bearings are of the self-aligning ring-lubricated type, divided horizontally and bolted together around the journals. The bearing is carried by a rib formed in the oil well, and is prevented from turning by a dowel in the top. The body of the turbine bearing is of cast iron, and that of the pump bearing is of composition; both being lined with white metal. The longitudinal position of the rotor is maintained by the thrust collars on the shaft which bear on each end of the pump bearing.

Oil wells are located below the bearings, as shown, and have covers, so that the bearing is completely enclosed. Each is fitted with a drain and sight glass.

The oil is prevented from leaking out around the shaft by the centrifugal action of the oil throws and thrust collars.

Turbine.—The rotor consists of a wheel secured to a hub which is keyed to the shaft; which also carries the rotors of the air and hotwell pumps. A groove, into which the buckets are fitted, is turned on the periphery of the wheel.

The buckets are made of cast bronze and are machined to true dimensions. The bucket consists of the steam channel, base and shroud piece. The base is inserted in the groove and held in place by rivets, as shown.

The nozzle and reversing chamber, which are made of cast bronze, are bolted into the lower part of the casing. The surfaces of the steam passages are highly finished to reduce friction. To prevent the leakage of steam, brass packing strips are fitted into grooves in the nozzle and reversing chamber; being held out by springs, so that a shoulder on the strip rests on another on the groove. Initially, the strips touch the rotor; subsequently, they are worn down to the exact running clearance, which is very small (about 0.005 inch). The clearance between rotor and the nozzle block, on each side, is about 0.015 inch.

A relief valve, set for 15 pounds, is located on the cover.

Governor (Fig. 6).—The spring is compressed between the adjusting nut and the spindle, which is connected by the knife-edge struts to the levers which carry the weights. Thus the

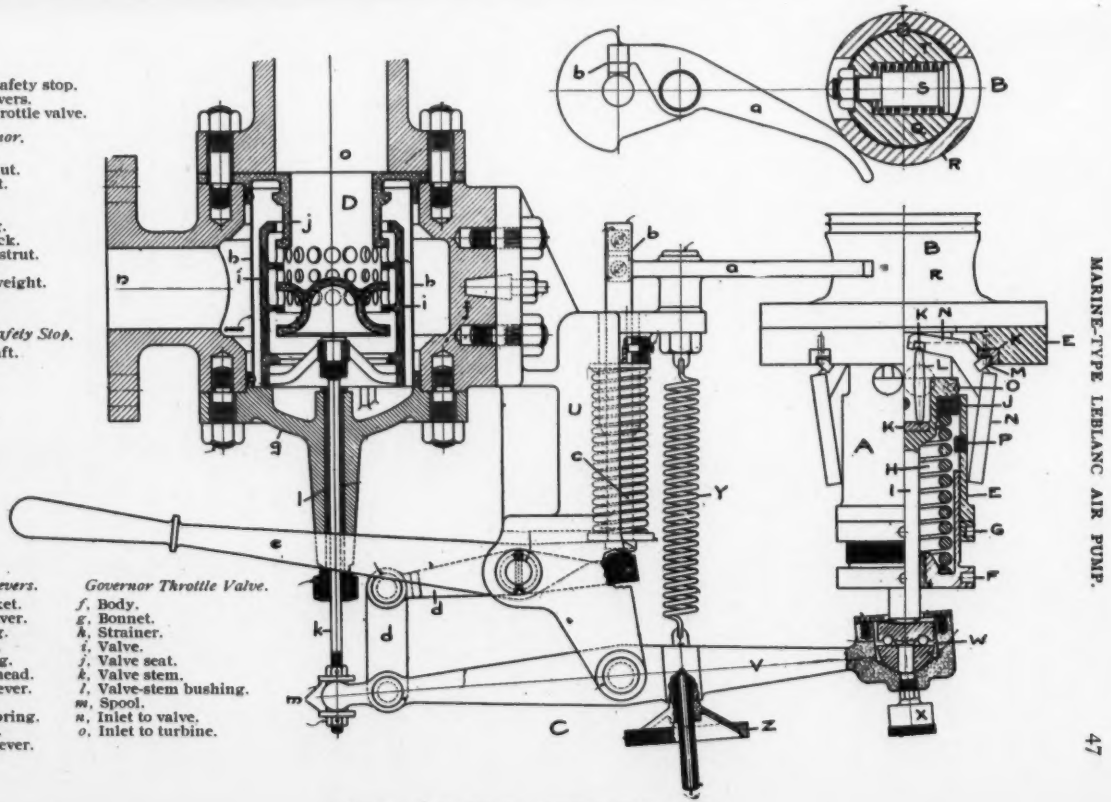
A, Governor.
B, Automatic safety stop.
C, Governor levers.
D, Governor throttle valve.

Governor.

E, Body.
F, Adjusting nut.
G, Locking nut.
H, Spring.
I, Spindle.
J, Ball bearing.
K, Bearing block.
L, Knife-edge strut.
M, Knife edge.
N, Lever and weight.
O, Roller.
P, Stop pin.

Automatic Safety Stop.

Q, Turbine shaft.
R, Body.
S, Stop pin.
T, Spring.



Governor Levers.

U, Lever bracket.
V, Governor lever.
W, Ball bearing.
X, Grease cup.
Y, Lever spring.
Z, Adjusting head.
a, Knock-off lever.
b, Latch.
c, Ram and spring.
d, Ram levers.
e, Engaging lever.

Governor Throttle Valve.

f, Body.
g, Bonnet.
h, Strainer.
i, Valve.
j, Valve seat.
k, Valve stem.
l, Valve-stem bushing.
m, Spool.
n, Inlet to valve.
o, Inlet to turbine.

Fig. 6.—GOVERNING MECHANISM.

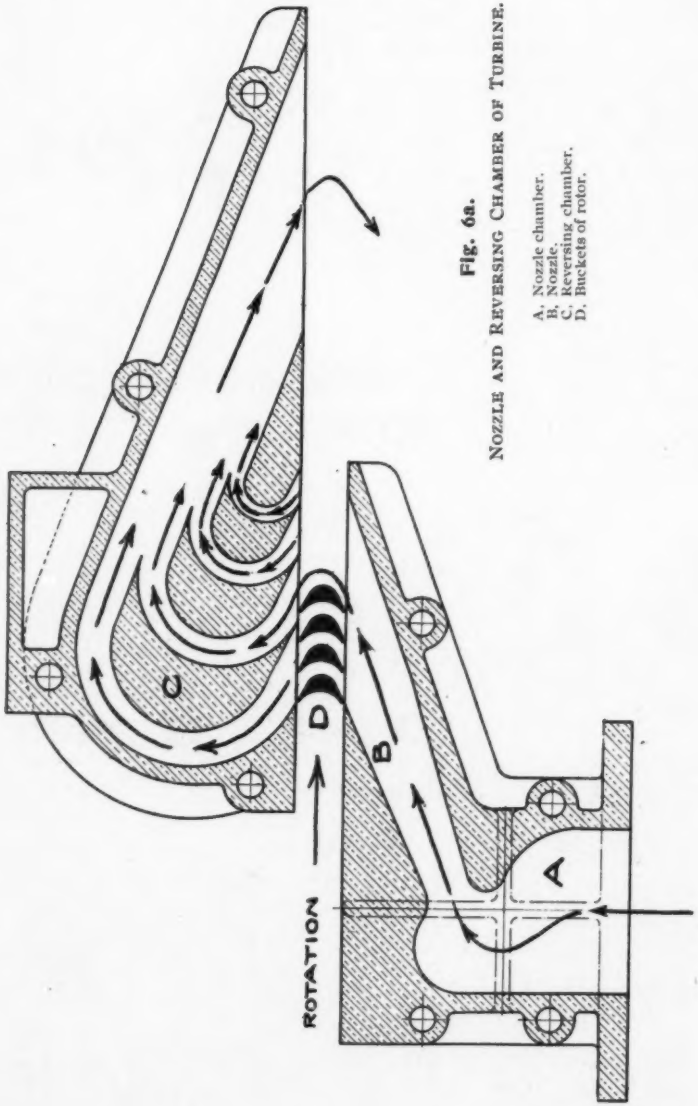


Fig. 6a.

NOZZLE AND REVERSING CHAMBER OF TURBINE.

A. Nozzle chamber.
B. Nozzle.
C. Reversing chamber.
D. Buckets of rotor.

force tending to cause the weights to move outward is resisted by the spring. The centrifugal force acting on the weights causes them to move outward, until it is balanced by the force of the spring. This movement of the weights is transmitted to the spindle, which is moved outward; thus causing the valve lever, which is held against the spindle by the lever spring, to move outward, opening the valve.

As the centrifugal force depends on the speed, and as this force is balanced by the spring; the governor can be set for the required speed by adjusting the compression of the spring. The force of the lever spring, which is transmitted to the spindle, also opposes the force of the weights; and, by adjusting its tension, the speed can be varied through a small range. The governor is adjusted for a speed of 2,200 r.p.m., and regulates within two per cent.

The governor consists of a body mounted on the end of the shaft, the cylindrical part of which contains the spring. The outer end of the latter rests on the adjusting nut, which is screwed into the body and secured with a locking nut; and the inner end rests on a ball bearing carried by the spindle. The spindle extends out through a brass bushing in the adjusting nut; forming a projection, against which, the lever bearing rests. In the recess in the inner part of the spindle, there is a steel block which forms the bearings for the knife-edge struts. The inner edges of the latter rest in bearing blocks inserted in the levers of the weights. The levers have inserted knife edges, which rest in bearing-block fulcrums inserted in the body. The knife edges and their bearing blocks are made of hardened steel to reduce friction.

The lower part of the spindle is fitted with rollers, which bear against the body to keep the spindle properly centered. The stop pin in the body restricts the outward movement of the spindle, and also that of the weights, to prevent the latter from striking the casing.

The lever is fitted with a ball bearing, which rests against the spindle; a grease cup being provided for lubrication. The valve end of the lever is fitted into a spool, which can be ad-

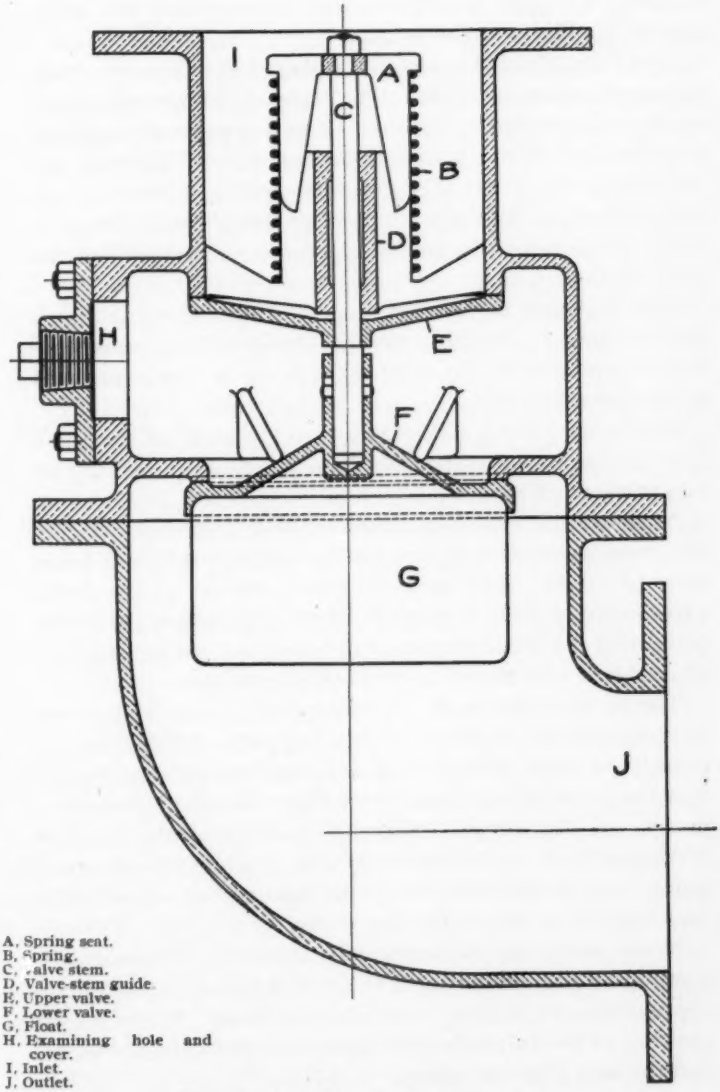
justed by the nuts at each end to give the desired position of the valve. The valve stem passes through a long bushing in the cover, a small clearance being provided to reduce the steam leakage. The valve, which is of the balanced type, surrounds the cage of the seat, as shown. A cylindrical strainer, made of perforated sheet iron, is fitted in the valve body outside of the valve.

The automatic safety stop consists of a pin, inserted radially in a hole in the shaft, which is held in its normal running position by a spring. The center of gravity of the pin is slightly eccentric to the center of the shaft; so that, in running, the centrifugal force tends to compress the spring, which is so loaded that this force will overcome that of the spring when the speed of the turbine is 10 per cent. above normal.

The outward movement of the pin increases the distance between the center of the shaft and its center of gravity, thus increasing the force acting upon it; so that, when once started, the pin moves out rapidly to the limit of its travel. In the latter position, its end protrudes sufficiently to strike the knock-off lever, which is moved outward, releasing the latch of the ram. The ram spring, which is held in compression by the latch; on being released, moves outward; forcing the ram against the ram levers, causing the valve to close.

The pin of the automatic safety stop remains out until the speed is reduced 10 per cent. below the normal, when the centrifugal force of the weight becomes less than the force of the spring, which causes the pin to return to its running position. After this, the governor can be put into operation again by moving in the ram (compressing its spring) with the engaging lever until the latch is engaged and the knock-off lever in its former position. The ram being removed from the ram levers, the valve lever is caused by its spring to return to its normal position with its bearing resting on the governor spindle, thus placing the governor in operation.

Air Pump (Fig. 5).—The water chamber is in two parts, one on eoch side of the rotor, centered in the casing around



A, Spring seat.
B, Spring.
C, Valve stem.
D, Valve-stem guide.
E, Upper valve.
F, Lower valve.
G, Float.
H, Examining hole and cover.
I, Inlet.
J, Outlet.

Fig. 7.—Air-Suction Check Valve.

the shaft, and each having a nozzle for directing the water against the blades of the rotor.

The rotor wheel is keyed to the shaft and carries the blading overhung on each side. The blades are of bronze, and are made by the extruding process. Riveting projections are cut on both ends of the blades, which are secured between two rings by riveting over the projections which extend through similar holes in the rings. The rings are riveted to the rotor wheel, as shown. The radial clearance on each side of the blades is 0.0625 inch.

The discharge nozzles are secured inside of the discharge pipe, as shown. Starting nozzles, fitted with a steam connection, are provided for starting the pump in case it is located above the water line.

Air-Suction Check Valve (Fig. 7).—The object of the check valve is to prevent salt water from accidentally backing up into the condenser.

As shown, there are two valves on a common stem. Under the lower valve there is a copper float having sufficient buoyancy to lift it. The stem fits into a sleeve on the lower valve, which is held in place by a pin; the slots in the sleeve permitting the small movement necessary for seating properly. The upper valve rests on a shoulder of the stem.

The spring, resting on the seat on the upper end of the stem, carries the weight of the moving parts, which compress it until the forces balance. In this condition, it may be said that the valve "floats;" as a very slight force will cause it to move. As the normal "floating" position of the valve is three-fourths of the full opening, and as an extremely small difference of pressure will cause the full opening; it will offer very little resistance to the flow of air.

If salt water should accidentally back up into the air-suction pipe, it will lift the float; thereby, reducing the weight on the spring, and causing the valve to close. In the closed position, as shown, the lower valve is held closed by the water, and the upper by the spring.

OPERATION.

As the condenser is connected with the sea through the air pump, it is of importance (1) to prevent sea water from entering the condenser and contaminating the feed water, and (2) to prevent the fresh water in the condenser from passing out through the air pump, resulting in the loss of feed water. If the pump is operated properly, there will be no danger of such accidents; but, as they are easily possible through improper operation, it is well to keep them in mind.

Not in Use.—When not in use, the turbine and air pump should be kept dry; the former to prevent corrosion, and the latter as a precaution against the leakage of salt water into the condenser.

The turbine steam and the exhaust valves should be closed. The drain to the bilge should be left open to keep the interior dry. The air-pump suction, discharge and gland-sealing water valves should be closed. The drains on the water and air suctions should be left open; the former to keep the air pump dry, and the latter to indicate and drain off accidental leakage.

The hotwell-pump discharge and gland-sealing water valves should be closed.

As there are no working parts that are likely to stick, it is not considered necessary to turn the rotor daily, and no means are provided for turning by hand. At least once a week, the pump should be turned by steam, and examined to see that it is secured properly. When the pump is turned by steam, both pumps should remain secured, as described above; except that water should be admitted to the glands for lubrication.

Starting.—In preparing to get underway, the air pump should be started at about the same time that an ordinary air pump would be. On account of the necessity of keeping the condenser well drained to avoid the loss of fresh water through the air pump, steam should not be admitted into the condenser until after the pump has been started.

With the pump secured as described above, proceed as follows:

Close air-pump drains. Open sealing water to pump glands. Open discharge valve of hotwell pump. Open turbine exhaust valve.

Open turbine stop valve, and bring up to speed. Close drain when turbine is clear of water.

Open the air-pump valves in the following order: (1) Sea discharge; (2) sea suction; (3) air suction.

Examine pump, and see that it is running properly, as described below.

In the above, the following should be noted: The gland water is turned on to lubricate the packing before starting. (1) The air pump is started dry; (2) the pump is put into operation; (3) the air suction to the condenser is opened. Thus unnecessary churning of the water is avoided, and there is no danger of the water backing up into the air-suction pipe.

Running.—As the regulation and lubrication are automatic, the pump requires practically no attention when running. It is advisable, however, to inspect it thoroughly once a watch to be sure that everything is working properly. In general, it should be noted that the bearings and pump glands are running cool; the oil reservoirs contain sufficient oil; the glands are properly sealed; the governor is working properly; the hotwell is drained; and the valves opened or closed as required.

The turbine stop valve should be opened wide, otherwise the governor will not function properly.

If the glands are not properly sealed, the leakage of air will impair the vacuum; which will also be the case if the air-pump drains are not closed tightly.

Pure mineral oil only should be used for filling the oil wells. Turbine oil should be used for the pump bearing; and for the turbine bearing, which runs comparatively hot, cylinder oil should be mixed with the turbine oil (about half and half) to reduce the fluidity of the latter.

Care should be taken to see that the condenser is properly drained; particularly after starting. If from any cause, such as the accidental closing of the discharge valve of the hot-

well pump, the condenser is not being drained properly; the condensed water will pass out through the air pump, resulting in the loss of fresh water. For this reason, the gage glass on the hotwell should be examined about once an hour.

As it is not considered necessary, no clearance gage is fitted on the turbine; and, while running, the clearance cannot be ascertained. Although there is little possibility of the rotor getting out of adjustment, it is advisable to listen for rubbing; particularly when starting.

If the pump gland is set up too tightly, the packing will wear rapidly. It should be set up "hand-tight," allowing the water to trickle out.

The working parts of the governor outside of the casing should be well lubricated.

The vacuum depends on the temperature of the discharge water, and the quantity of air entering the condenser. The ideal vacua, the highest attainable, correspond to temperatures of discharge as follows: 100°, 28.08; 90°, 28.59; 80°, 28.98; 70°, 29.27, and 60°, 29.48. (Vacua referred to a barometer of 30 inches.) Where the air leakage is moderate, the pump will maintain practically the ideal vacuum; and this will be reduced according to the increase of the leakage.

In general, the discharge temperature should not exceed 90 degrees in turbine ships. The air leaks should be reduced to a minimum; and, with this end in view, the turbine glands; which, if not properly sealed, permit large leakage; should receive careful attention.

The vacuum gages should be correct to about one-tenth of an inch; and, where possible, a mercury column should be used. If the latter is not practicable, as on small, lively ships; it should be used for checking the working gages. Absolute and spring gages, particularly the latter, are often incorrect; and, unless they are checked frequently, the vacuum recorded will probably differ considerably from the actual vacuum.

Shutting down.—In turbine ships, it is usual to keep the air pumps running for a sufficient length of time after the

main turbines have been shut down to thoroughly dry them out. The procedure is as follows:

Close the air-suction, sea-suction and sea-discharge valves of the air pump; and open both drains.

Close turbine stop and exhaust valves, and open drain. Close discharge valve of hotwell pump, and shut off gland-sealing water of both pumps.

Overhauling, etc.—About once a quarter, the covers should be lifted for internal examination. Before the rotor can be lifted; the turbine and air-pump covers, the entire hotwell pump, and the bearing covers must be removed. The rotor can be lifted out by hand, as shown in Fig. 8; care being taken not to injure the governor, and the buckets of the turbine and air-pump rotors.

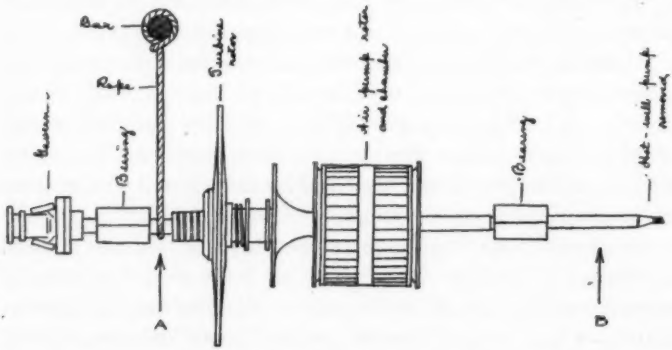


Fig. 8.—METHOD OF LIFTING ROTOR.

While open, care should be taken to prevent foreign substances from getting into the casing; and, when no work is being done, it should be covered.

The longitudinal and radial clearances should be measured; the former by inserting feelers between the rotor and nozzle blocks of the turbine, and the latter by measuring the distance between the lower surface of the shaft and convenient points on the casing.

If necessary, the longitudinal clearance can be adjusted by

adjusting the thrust collars; which is done by inserting or removing the liners provided for this purpose. If the position of the rotor is too low, as may be caused by wear of the bearings; the radial clearance may be adjusted by placing liners under the bearings, or by putting in new bearings. It is probable, however, that the pump will run for a long time before requiring such adjustments.

The float (play in the thrust bearing) should be small to insure the accurate operation of the governor, and prevent a large working clearance between the rotor of the turbine and the packing strips of the nozzle block. If the latter wear down too much, they should be renewed. The float should not exceed 0.004 inch.

The vacuum joints of the pump should be perfectly tight (this also applies to all joints subjected to vacuum); and, with this end in view, all parts through which air can leak should be coated with paint—over the edge of the joint, around bolt heads, nuts, etc. The coating should be fairly thick, and should have an unbroken surface. With joints sealed in this way, it is possible to tell by examination whether they are tight; which is not possible otherwise. For this purpose, asphaltum or machine enamel paint is recommended.

As the blades of the turbine and air-pump rotors have sharp edges, they are comparatively frail; and care should be taken to prevent injury by rubbing, or by foreign substances. The strainers which are fitted should be kept in efficient condition. If the blades are slightly bent or nicked, they should be straightened and made smooth.

Square flax packing may be used in the pump glands if necessary. When the packing is replaced, it should be well covered with graphite; which should also be used on the packing rings of the turbine gland.

The governor should be tested; and, if necessary, adjusted to give the designed speed of 2,200 r.p.m. This can be done by taking the speed from the hotwell end of the shaft, after having removed the plug provided for this purpose. A small adjustment of speed can be made with the lever spring; and

further adjustment with the adjusting nut of the governor. The required position of the valve can be obtained by adjusting the spool.

Occasionally, the oil wells should be cleaned and filled with fresh oil.

Spare Parts.—The following spare parts should be carried for duplicate pumps: one rotor complete (including bearings, packing rings, governor, etc.), one set of packing strips and springs for nozzle blocks, and one set of packing for each gland of one pump.

Performance.—The probable performance of the marine-type Leblanc air pump given in Table I was assumed from numerous tests made by the Westinghouse Machine Company; and represents average good results for a pump of that capacity, which is one-third larger than the standard practice of a European firm, which has had considerable experience and success with the Leblanc pump. It is probable that the smaller capacity will be ample for naval work; and, because of its better economy, it would be more desirable.

Table I also gives the performance of the air-pump outfits on the *Chester* and *Salem;* the former being of the augmenter, and the latter of the dry-air type. The data for these outfits were obtained from the comparative trials of the scout cruisers, published in the J. A. S. N. E., Vol. XXII, No. 3.

As the data were insufficient, certain values being assumed; the comparisons are only approximate; and it may also be noted that the figures vary according to the capacities of the pumps. However, it is believed that the figures given in Table II are fairly close approximations to the results that will be obtained with units of large power.

For purposes of comparison, it was assumed that the air leakages for all the tests were no greater than those which would obtain in good turbine practice.

In making a comparison of air-pump outfits, with the object of determining which will require the least steam for both the air pump and the main turbine to give the required full power of the turbine; it is necessary to know the increase in the

steam consumption of the turbine caused by the inefficiency of the air pump in not maintaining the ideal vacuum. The effect of vacuum on the turbine water rate, both ideal and actual, is shown in Fig. 9. From the curve per cent. increase in water rate per inch loss of vacuum, it will be seen that the loss between 28 and 29 inches of vacuum is about 4 per cent.; and, as other tests have given from 3 to 5 per cent., depending on the efficiency of the turbine, the probable value for the cases under consideration may be taken as 4 per cent.

It may be noted that the loss due to vacuum depends on the design and efficiency of the turbine; and that it is less with one designed for low vacuum than with one designed for high vacuum. And also, for low powers; the loss due to vacuum is greater than at full power. However, to obtain a correct comparison; it is necessary to use the same loss in all the cases under consideration; and, as these turbines were designed for approximately the same conditions, it is probable that 4 per cent. is not far in error.

In determining the economy of an air-pump outfit, as stated above; the increase in the steam consumption of the main turbine due to the inefficiency of the air pump should be considered, as well as the steam consumed by the air-pump outfit. The reason for this is obvious; for, if only the steam consumed by the air-pump outfit is considered, the important factor of air-pump efficiency and the resulting increase in the steam used by the main turbines is neglected.

For example, if the ideal vacuum is maintained; the steam used by the turbine will be the minimum; and, as there is no loss due to the inefficiency of the air pump, its economy will be represented by the steam used by the air pump only.

If, however, the vacuum is less than the ideal; the turbine will require an additional quantity of steam to produce the same power; for which, the inefficiency of the air pump is responsible. In this case, the economy of the air pump will be represented by the additional steam required by the turbine in addition to its own steam.

The steam chargeable to the air-pump outfits given in the

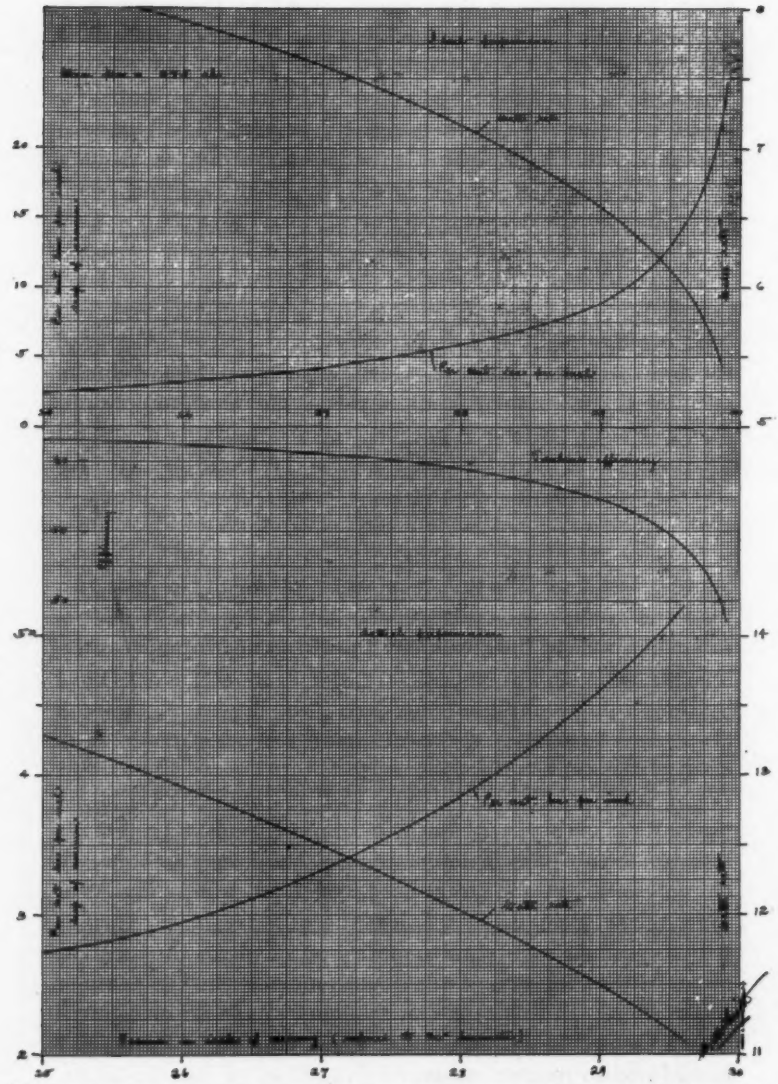


Fig. 9.—Curves Showing the Effect of Vacuum on the Turbine Water-Rate.

tables was determined by this method, which gives what may be called the "over-all economy."

The results obtained on the *Chester* and *Salem* do not agree very closely with the average good results, as shown by a number of tests of similar outfits; on the *Chester*, the results are not as good; and, on the *Salem*, they are better.

With the augmenter type, the terminal difference is usually from 4 to 6 degrees F.; and 5 may be taken as a good average.

With the dry-air type, the terminal difference is usually from 7 to 10 degrees F.; and 8 may be taken as a good average.

With the Leblanc pump, the ideal vacuum is being maintained in several plants; and 5 degrees terminal difference, which is seldom exceeded, may be taken as a conservative value.

If we assume the terminal differences given above, and the smaller size Leblanc pump (two-thirds of the size given in Table I); the comparison will be as given in Table II.

It should be noted that, as the weight of the Leblanc pump is not proportional to the turbine power, the figures given do not cover all cases; but they are approximately correct for large powers. In this connection, it should be noted that the water used by the turbines of the *Chester* and *Salem* is respectively 2.8 and 2.5 times that for the Leblanc; and that a Leblanc pump for the same power would be somewhat lighter per 1,000 pounds of steam used by the turbines. If the smaller-capacity pump were used, the weight would be still further reduced; but only slightly.

As shown in Table I, the weights of the air-pump outfits of the *Chester* and *Salem* are respectively 4.5 and 8 times that of the Leblanc: a decided advantage for the latter in naval work, where weight is an important consideration.

The steam chargeable to the air pump on the *Chester* and *Salem* is respectively 1.32 and .795 times that of the Leblanc. However, it is probable that the values in Table II are nearer correct; and that the augmenter and dry-air types are respectively 1.46 and 1.2 times the Leblanc.

The exhaust steam from the air-pump outfit which passes into the auxiliary-exhaust system may be utilized for feed heating, or for developing power in the low-pressure stages of the main turbine.

On the *Chester*, the air pumps exhaust into the auxiliary-exhaust line, and the augmenters into the augmenter condensers; so that, the steam available for feed heating, that from the air pumps only, was 5,500 pounds per hour, or 71 per cent. of the total steam used by the air-pump outfits.

On the *Salem*, the dry-air pumps exhaust into the auxiliary-exhaust line, and the hotwell pumps into the main condensers; the steam for feed heating being that from the dry-air pumps 2,585 pounds per hour, or 61 per cent.

With the Leblanc pump, all the steam used is available for feed heating; and, consequently, a thermal comparison would be more in its favor.

It should also be noted that the back pressure on the auxiliary-exhaust lines of the *Chester* and *Salem* was 7 pounds, whereas the exhaust pressure for the Leblanc is 10 pounds.

In general, it may be said that the augmenter and Leblanc types will give about the same vacuum efficiency, but that the augmenter outfit will require more steam; also, that the dry-air type will require less steam than the Leblanc, but that the vacuum efficiency will be less; and also, that the over-all economy of the Leblanc will be better than either.

TABLE I.—PERFORMANCE OF THE LEBLANC AIR PUMP AND THE AIR-PUMP OUTFITS OF THE "CHESTER" AND "SALEM."

	Chester.	*Salem.*	Leblanc.
(1) Temperature of discharge water, degrees F.	77.	87.	80.
(2) Temperature corresponding to air-pump vacuum, degrees F.	87.	92.	85.
(3) Temperature difference, degrees F.	10.	5.	5.
(4) Air-pump vacuum, inches of mercury	28.70	28.50	28.80
(5) Ideal vacuum (corresponding to discharge temperature), inches of mercury	29.07	28.72	28.98
(6) Difference between ideal and actual vacuum-inches of mercury	.37	.22	.18
(7) Per cent. of ideal vacuum [(4)÷(5)×100]	98.73	99.23	99.38

	Chester.	*Salem.*	Leblanc.
(8) Steam used by main turbines, lbs. per hr...	282,000	253,000	100,000
(9) Per cent. loss due to 1-inch drop of vacuum.	4	4	4
(10) Steam for main turbines that could be saved by maintaining ideal vacuum, lbs. per hr., [(9)×(6)×(8)]	4,180	2,230	720
(11) Steam used by air pump, lbs. per hr.	7,791	4,247	2,500
(12) Steam used by air pump, plus steam that could be saved by maintaining the ideal vacuum. Total steam chargeable to air pump, pounds per hour	11,971	6,477	3,220
(13) Steam used by air pump per 1,000 pounds used by main turbines, pounds	27.6	16.75	25.0
(14) Steam chargeable to air pump per 1,000 pounds used by main turbines, lbs.	42.5	25.6	32.2
(15) Same referred to Leblanc as unity	1.32	.795	1.00
(16) Weight of air-pump outfit, pounds	31,834	50,882	2,500
(17) Weight of air-pump outfit per 1,000 pounds of steam used by main turbines, pounds	113	201	25
(18) Same referred to Leblanc as unity	4.52	8.05	1.00

TABLE II.—PERFORMANCE OF AIR-PUMP OUTFITS OF THE LEBLANC, AUGMENTER AND DRY-AIR TYPES.

	Augmenter.	Dry-air.	Leblanc.*
(1) Terminal difference, degrees F	5	8	5
(2) Steam chargeable to air pump per 1,000 pounds used by main turbines, pounds	34.8	28.7	23.8
(3) Same referred to Leblanc as unity	1.46	1.20	1.00

TABLE III.—ADDITIONAL DATA FOR THE LEBLANC AIR-PUMP GIVEN IN TABLE I.

(1) S.H.P.	50
(2) Steam consumption, pounds per S.H.P. per hour	50
(3) Volumetric displacement, cubic feet per second	10
(4) Volumetric displacement per pound of steam used by main turbines, cubic feet	.36
(5) R.P.M.	2,200
(6) S.H.P. of main turbines	8,000
(7) Steam consumption per S.H.P. of main turbines, lbs. per hr.	.312
(8) Steam chargeable to air pump per S.H.P. of main turbines, lbs. per hr.	.4025
(9) Weight per S.H.P. of main turbines, pounds	.312
(10) Turbine-nozzle pressure, pounds per square inch, gage	175
(11) Exhaust back pressure, pounds per square inch, gage	10

* Pump one-third smaller than the one given in Table I.

AMERICAN PRACTICE IN USING SUPERHEATED STEAM.

By Captain Clarence A. Carr, U. S. Navy.

During the summer of 1910 the Bureau of Steam Engineering, Navy Department, desired information as to the best American practice in stationary power plants using superheated steam. In order to get this information the Bureau's various inspectors of machinery and engineering material were directed to submit reports on the use of superheated steam in their districts, or in cases which might have come under their personal observation. By this means a large quantity of information was collected which included letters giving the experience of many well-known stationary engineers who are known to have had wide experience in the use of superheated steam. These reports determine the steam pressure and degree of superheat in daily use with steam turbines and with reciprocating engines in a great many well-designed power plants. The difficulties which have been encountered in the use of superheated steam are also given with the methods by which these difficulties have been overcome. It appears that superheated steam is very extensively and successfully used by corporations such as street railways and electric-light companies, whose managers must realize the general public dissatisfaction with the company which would be caused by any failure to supply power as required. This is an evidence that superheated-steam installations can be made reliable, and that these large corporations have been satisfied that the gain in economy from the use of superheated steam, is sufficient to warrant the increased cost of an installation which will not be liable to break down.

Reprints of this article can be had of the Secretary-Treasurer of the Society for forty cents each, postpaid.

The data that has been collected shows that there is very little difference of opinion among stationary engineers, who are familiar with the use of superheated steam, as to the conditions which should govern its use in order to get the most satisfactory results. In fact, the opinions given agree so closely, and several of the letters which have been received treat the subject so completely, that one of these letters might have been quoted as giving the best American practice on the subject of the use of superheated steam. It is, however, believed that the method which has been followed of quoting extracts from letters will serve to bring out many minor points more clearly. The opinions do not agree as to the actual economy to be expected under different degrees of superheat, but generally do agree as to the advisability of the use of superheated steam. The statements that if designs take into account the probable variations in temperature, that for total temperatures up to about 500 degrees, superheated steam should give no more trouble than saturated steam, are particularly important, and are shown to be correct as regards stationary power plants.

All quotations made have been revised by the original writers in order that there should be no possibility of error through quoting part of a letter. Descriptions have also been given in considerable detail of the difficulties encountered in several of the first installations of superheaters in large power plants in this country.

All pressures are given per gage and all temperatures in degrees Fahrenheit.

The Edison Electric Illuminating Co., Boston, Mass.

Information from Mr. I. E. Moulthrop, Mechanical Engineer.

"Thirty-six Babcock and Wilcox boilers, 110 square feet grate surface each. Turbine installation of vertical Curtis turbines. Total capacity, 51,000 kw., divided as follows: Three turbines of 12,000 kw. each, maximum rating; two turbines of 7,500 kw. each, maximum rating. Steam pressure

about 200 pounds per gage. Superheat 150 degrees under normal conditions.

"No difficulties at all have been experienced with the superheaters themselves. At this station the standard construction is to make all pipe fittings of cast steel. Valves, except throttle valves, are gate valves. Valves are made with cast-steel bodies, phosphor-bronze discs and nickel-bronze seats. Pipe joints are made with Klingerit and also with special McKim gaskets for superheat. The auxiliaries are mostly reciprocating, and have given no trouble which could in any way be attributed to the use of superheated steam. At this plant the use of superheated steam has been found economical and satisfactory in every way, and the installation of additional boilers without superheaters would not be considered. From the experience of this company with superheated steam, which covers a period of about 12 years, it is believed that with not over 125 degrees superheat, first-class steam valves of standard make will give but little more trouble than they would with saturated steam. It goes without saying that where superheated steam is to be used the steam piping should be designed with this in view. Superheated steam causes increased expansion, and it is readily seen that poorly laid out steam lines may give much trouble on account of the increased amount of expansion and contraction over that due to saturated steam.

"This was one of the first companies to use superheated steam on a large scale in the United States, and while some of the difficulties we encountered were very trying for the time being, they were really not very serious. Our first installation of high-pressure steam piping had valve bodies and fittings made of air-furnace gun iron. This metal was used on account of its superiority in tensile strength and ductility, and because at that time it was not possible to have these various fittings made in cast steel. We later on found some fittings which showed distress, and at that time it was attributed to the use of gun iron in connection with superheated steam. The first gate valves installed were of a compara-

tively new design, having gun-iron bodies with seats cast in the bodies, and the plugs being of the same material without any facing material. These seats did not stand up very well under superheated steam. They blistered and were removed, valves of the type already described being installed.

"The first installation of high-pressure steam piping had flange joints made up with the ordinary corrugated copper gaskets. There was some trouble from joints leaking from the gaskets becoming brittle and cracking. This has been attributed to superheated steam, but I believe this opinion to be an error, and would have no hesitation in using corrugated copper gaskets, suitably annealed, in flange joints of superheated steam lines where the total temperature did not exceed 650 degrees to 700 degrees F."

The Edison Illuminating Company, Detroit, Michigan.

Information from Mr. Alex. Dow, Vice-President and General Manager, Member A. S. M. E.

"Steam pressure, 210 pounds per gage at boiler side of superheater; at turbines, 180 to 190 pounds Normal superheat, 200 degrees in power house No. 1 and 150 degrees in power house No. 2. This degree of superheat is exceeded when the boilers are forced for any reason. The reduction of superheating surface in No. 1 power house is now being considered, but this will not be done until the increased erosion of turbine vanes due to this reduction of superheat has been fully investigated. The superheaters which were originally installed gave trouble by the tube ends drawing out of the tube sheets. This was remedied by the use of ferrules and by expanding the tube ends inside the sheet. The piping in this plant was originally fitted with cast-iron and semi-steel valves and fittings. After they had been in use for some time they became badly distorted and in some cases honeycombed and cracked. Some pipe fittings increased in length more than $\frac{1}{4}$ inch. The turbines were originally equipped with electrically-controlled nozzle valves, which gave trouble by sticking until the casings and valves had been changed to

cast steel. This sticking was caused by the valves and casings warping under high temperature. This power house was designed to use saturated steam for all auxiliaries, and it had been assumed that no probable superheat would be detrimental to the main steam turbines. In practice, during the early operation of the plant, it was found that one or more turbines might be run with a very light load while the boilers in service were being forced to nearly the full capacity in order to supply the auxiliaries, heating plant, etc., which used saturated steam. Under these conditions less than the usual proportion of steam passed through the superheaters and the actual temperatures of superheated steam were sometimes in excess of 700 degrees F. These conditions, doubtless, aggravated the trouble with cast-iron valves and fittings. The trouble has been remedied by substituting annealed cast-steel valves and fittings throughout. In No. 2 power house cast-steel valves and fittings were used from the start, the valves having seats and discs of a nickel alloy. These have given no trouble whatever under the degree of superheat used, 150 degrees. No special trouble has been found with packing for valve stems. Copper gaskets were at first used for pipe joints, but they did not last long. They have been replaced by soft-iron gaskets with satisfactory results. In No. 2 power house the auxiliaries are served from superheated steam mains, but steam-turbine or motor-driven auxiliaries are used with the exception of two Laidlow-Dunn-Gordon dry-air pumps and two engine-driven circulating pumps. The turbine-driven auxiliaries operate satisfactorily, so far as superheat, or absence thereof, affects their operations.

"The valves used on our superheated-steam lines deserve special notice. These valves are made in England, by J. Hopkinson & Co., Limited, Huddersfield, and are known as the Hopkinson-Ferranti type. The gate area in these valves is restricted to approximately one-quarter the area of the pipe, while at the flanges it has the full area. The reduction and restoration of area are effected by shaping the valve passage as a Venturi tube, the valve gate and seats being at the point

of least area. The effect is increased velocity of steam with reduction of pressure, and subsequent reduced velocity with increasing pressure. With steam-pipe area proportioned according to the usual practice, the net loss of pressure from one of these valves is very small, but with the use of high steam velocities in pipes allowed in most recent design, the loss is appreciable. The gains by the use of this device are less chance for distortion of seat and gate; the deflection of the steam away from the seats so that cutting is reduced or prevented; also less friction in opening and closing valves, thereby avoiding the necessity of a by-pass. A detail of this valve is the use of a ring to bridge the distance between seats when the valve is open.

"American manufacturers have lately offered valves coned to a less degree than the valves made in England, and therefore better suited to high steam-pipe volicities."

Commonwealth Edison Company, Chicago, Ill.

Information from Mr. W. L. Abbott, Chief Operating Engineer for the Company.

"The tendency in this company toward higher boiler-pressures is shown by conditions found in its steam-turbine power houses. In 1903 our Fisk Street power house was put in service, using steam at 180 pounds pressure, and three years later additional units were installed using steam at 200 pounds pressure. In 1909 our Quarry Street Station was put in service, using steam at 210 pounds pressure, and we are now planning a third power house, to be known as the Northwest Station, which will use steam at 250 pounds pressure. The degree of superheat in the Fisk Street and the Quarry Street power houses averages about 150 degrees, but as this temperature often varies 50 degrees either way from the normal. The tendency is to cut down the normal temperature in order to avoid excessive temperatures at the upper limit. In the new Northwest Station we have specified that the normal superheat shall be 125 degrees F.

"Careful tests have recently been made which indicate that

cast steel. This sticking was caused by the valves and casings warping under high temperature. This power house was designed to use saturated steam for all auxiliaries, and it had been assumed that no probable superheat would be detrimental to the main steam turbines. In practice, during the early operation of the plant, it was found that one or more turbines might be run with a very light load while the boilers in service were being forced to nearly the full capacity in order to supply the auxiliaries, heating plant, etc., which used saturated steam. Under these conditions less than the usual proportion of steam passed through the superheaters and the actual temperatures of superheated steam were sometimes in excess of 700 degrees F. These conditions, doubtless, aggravated the trouble with cast-iron valves and fittings. The trouble has been remedied by substituting annealed cast-steel valves and fittings throughout. In No. 2 power house cast-steel valves and fittings were used from the start, the valves having seats and discs of a nickel alloy. These have given no trouble whatever under the degree of superheat used, 150 degrees. No special trouble has been found with packing for valve stems. Copper gaskets were at first used for pipe joints, but they did not last long. They have been replaced by soft-iron gaskets with satisfactory results. In No. 2 power house the auxiliaries are served from superheated steam mains, but steam-turbine or motor-driven auxiliaries are used with the exception of two Laidlow-Dunn-Gordon dry-air pumps and two engine-driven circulating pumps. The turbine-driven auxiliaries operate satisfactorily, so far as superheat, or absence thereof, affects their operations.

"The valves used on our superheated-steam lines deserve special notice. These valves are made in England, by J. Hopkinson & Co., Limited, Huddersfield, and are known as the Hopkinson-Ferranti type. The gate area in these valves is restricted to approximately one-quarter the area of the pipe, while at the flanges it has the full area. The reduction and restoration of area are effected by shaping the valve passage as a Venturi tube, the valve gate and seats being at the point

of least area. The effect is increased velocity of steam with reduction of pressure, and subsequent reduced velocity with increasing pressure. With steam-pipe area proportioned according to the usual practice, the net loss of pressure from one of these valves is very small, but with the use of high steam velocities in pipes allowed in most recent design, the loss is appreciable. The gains by the use of this device are less chance for distortion of seat and gate; the deflection of the steam away from the seats so that cutting is reduced or prevented; also less friction in opening and closing valves, thereby avoiding the necessity of a by-pass. A detail of this valve is the use of a ring to bridge the distance between seats when the valve is open.

"American manufacturers have lately offered valves coned to a less degree than the valves made in England, and therefore better suited to high steam-pipe volicities."

Commonwealth Edison Company, Chicago, Ill.

Information from Mr. W. L. Abbott, Chief Operating Engineer for the Company.

"The tendency in this company toward higher boiler-pressures is shown by conditions found in its steam-turbine power houses. In 1903 our Fisk Street power house was put in service, using steam at 180 pounds pressure, and three years later additional units were installed using steam at 200 pounds pressure. In 1909 our Quarry Street Station was put in service, using steam at 210 pounds pressure, and we are now planning a third power house, to be known as the Northwest Station, which will use steam at 250 pounds pressure. The degree of superheat in the Fisk Street and the Quarry Street power houses averages about 150 degrees, but as this temperature often varies 50 degrees either way from the normal. The tendency is to cut down the normal temperature in order to avoid excessive temperatures at the upper limit. In the new Northwest Station we have specified that the normal superheat shall be 125 degrees F.

"Careful tests have recently been made which indicate that

the steam economy was improved about three per cent. by an increase of superheat from 100 degrees to 150 degrees, but it is doubtful if this increase in fuel economy fully compensates for all of the expense and annoyances attendant upon its production and control. We have, however, found a high degree of superheat necessary to prevent the excessive erosion of turbine blades by the condensation which would otherwise take place in the expansion of steam through the turbine from a gage pressure of 200 pounds to an absolute pressure as low as one-half inch of mercury. We have found, in fact, that in the case of some of our turbines which, on account of a change in boiler baffling are regularly supplied with steam at a superheat temperature of 100 degrees, the bucket erosion is so much greater than in the case of those turbines which receive steam at a temperature 50 degrees higher, that the lower superheat is a matter of serious concern.

"The disadvantages encountered in the use of superheated steam result from the expansion and contraction which takes place in the superheater and in the steam line and its fittings, and from the effect of the high temperature on the metals used. After a considerable amount of costly experience we have adopted cast-steel as the material to be used for flanges and fittings and for valve bodies. For valve discs and seats we use nickel-steel."

Interborough Rapid Transit Company, New York.

Information from Mr. H. G. Stott, Superintendent of Motive Power.

"Steam pressure, about 200 pounds per gage; superheat, not over 150 degrees. Our experience is that the application of 100 degrees superheat to the ordinary standard high-pressure valves and fittings has resulted in such distortion and cracks as to necessitate their entire renewal. The most satisfactory material for use in valves, valve seats and valve discs we have found to be cast-steel for bodies and Monel metal for seats and discs. For steam joints in piping, ground joints have been found the most satisfactory. Any degree of super-

heat above 100 degrees has a bad effect on cylinders and valves, unless they are especially designed for its use; but with a turbine plant, 50 to 75 degrees higher superheat can be safely carried.

"It is my opinion that there is a tendency to exaggerate the net value of the use of superheated steam and that the most important gains are really to be made by the use of a moderate degree of superheat, such as 75 degrees for the cylinder of an engine and 125 degrees to 150 degrees for a steam turbine. Above these points the increased original cost and the cost of maintenance is likely to diminish the saving very materially. A reaction has set in against the use of superheat above 150 degrees F., and most new plants specify 125 degrees at throttles for turbines and 75 degrees at throttles for engines. This means that a total maximum temperature of steam above 550 degrees is not considered a good investment.

"With new engines I consider it desirable to progressively increase the degree of superheat, but with a new boiler installation no harm will result from using the maximum degree of superheat and temperature at once, provided all steel construction is used."

The New York Edison Company, New York City.

Information from Mr. J. P. Sparrow, Chief Engineer.

"Steam pressure, 190 to 200 pounds per gage; superheat, 100 degrees under normal conditions. We have had no difficulties in keeping our superheaters in an efficient condition. They are all of the Babcock & Wilcox type and built into the boiler setting. We have not found it difficult to keep valves tight when they are made of proper material, and the results obtained from our type of valve, which is an open-hearth steel body with nickel-steel seats and seat rings, have been entirely satisfactory. It has been found more or less difficult to keep packing tight, especially with valves that are operated frequently. My own preference is for a molded Vulcabeston packing ring.

"Steam turbines demand for efficiency of operation and economy of maintenance higher temperatures than have been found desirable in reciprocating engines. The moderate degrees of superheat which we have used, viz: 100 degrees to 150 degrees, have introduced no difficulties which have been found objectionable, and with the exception of demanding especial attention in the matter of lubrication for the small reciprocating unit, I can see no appreciable difference between the operation of superheated steam plants and saturated steam plants within the limits specified. We do not use superheated steam on our main engines. On the small auxiliary engines there is more or less wear in the cylinders and valve chests and they require constant attention to keep them in efficient condition. Our practice is to use turbine-driven auxiliaries as far as possible; with which there is, of course, no trouble from superheat temperatures. We do not, however, have sufficient trouble with the small reciprocating auxiliaries to condemn their use."

Public Service Electric Company, Newark, N. J.

Information given by Mr. J. T. Whittlesey, Chief Engineer.

"It is the practice of this company to use superheated steam in all its steam-turbine plants; steam pressure, 190 to 200 pounds per gage; degree of superheat, not over 150 degrees. In most cases where records of temperatures are available we find that we are getting about 125 degrees superheat at the turbines. For steam piping Vanstone or welded flanges are used for all joints, and cast-steel for fittings and valves. The most satisfactory gasket which has been found is of thin molded Vulcabeston. The piping in our Marion Power Station, Jersey City, N. J., has been in use for five years under superheated steam of from 125 degrees to 150 degrees F., and from 190 to 200 pounds gage pressure, with entirely satisfactory results. No fittings or valves have broken and the joints are still tight with the original gaskets. We are running a few turbines on saturated steam in stations where we already

have reciprocating engines. Up to the present time most of our auxiliaries are running on saturated steam which has proven very satisfactory, and it has eliminated the usual troubles experienced with packing and valves where superheat is used on small apparatus. I would add, however, that I am now designing a new station where I will have no saturated steam. Practically all the auxiliaries will be turbine-driven. However, when especially built for that service I think there is no difficulty with operating almost any pump under a reasonable amount of superheat, say 125 degrees."

Omaha Electric Light and Power Company, Omaha, Neb.

Information given by Mr. A. C. Anderson, Superintendent of the plant.

"Steam pressure, 200 pounds per gage; superheat, 150 degrees F., under normal conditions. This company operated for about one year under 200 degrees of superheat, but from observation of the installation it was considered unsafe to use steam of this temperature, and the degree of superheat was cut down to 150 degrees F. One 10-inch cast-iron gate valve was found to have increased in length over $\frac{1}{4}$ inch. A test specimen from an 18-inch tee was tested and showed a loss of 45 per cent. of its original tensile strength. The new installation of steam-piping is of extra-heavy steam pipe with wrought-steel flanges welded on. All tees, valves, expansion joints and other fittings are of cast steel. Gate valves are used with cast-steel bodies and discs and nickel-steel stems. Leslie reducing valves of bronze are used and have been found satisfactory. The new installation has shown no ill effects from the use of superheated steam. It is found that no more coal is burned running with superheated steam than with saturated, but when using superheated steam the turbines will carry an increased load of at least 10 per cent. A series of tests have also been conducted with degrees of superheat varying from 50 degrees to 150 degrees, and all things being considered it was found that 150 degrees gave the most economical results. There has been no difficulty at all with the superheaters themselves."

Cambria Steel Company, Johnstown, Pa.

Information from Mr. John W. Gocher, Chief Engineer.

" Steam pressure, 150 pounds per gage; superheat, 100 degrees. In some of the power plants steam is carried at 125 pounds, but the superheaters are so proportioned as to give 100 degrees superheat under normal conditions. The degree of superheat varies from 80 to 140 degrees F., depending principally on the rate of driving the boilers. This company has boilers rated at 58,000 H.P., and of these about 14,000 H.P. are supplied with superheaters. All new boilers which are installed are equipped with superheaters, and, as it becomes possible to do so, old installations are equipped with superheaters if the condition of the boilers warrants the expenditure. There has been very little trouble with superheaters. In one case superheater tubes sagged, and this was corrected by a change in the support for the tubes. Also one superheater, in a battery of twenty boilers of 300-H.P. each, became scaled up so that it was necessary to remove the tubes. The feed water for this battery of boilers was very hard, but no reason has been found for the excessive scaling of this particular boiler. Gate valves are generally used, and there has been no trouble with them or the valve seats of the safety valves. Standard high-pressure cast-iron valves were originally used, and no difficulty has been experienced with these valves, a great many of them being still in service and having been in service for four years or more. Repeated measurements of cast-iron elbows and tees in superheated steam mains have failed to show any perceptible growth in any casting. In our latest specifications for gate valves for superheated steam, semi-steel bodies with nickel-steel valve stems and nickel-composition seats and discs have been specified. This change has not been made on account of any trouble that cast iron has given here, but on account of trouble which it has given in other places.

" Our boilers were equipped with superheaters, not so much with the idea of obtaining superheated steam for use in our various prime movers as with the idea of eliminating the

troubles due to condensation in our comparatively long steam mains. In some cases these mains are so long that we obtain at the point of use practically dry steam, but with no superheat. With the moderate degree of superheat used we have no trouble, whether used in engines or elsewhere."

Old Colony Street Railway, Quincy Point, Mass.

"Steam pressure, 175 to 200 pounds per gage; normal superheat, 100 degrees. The following data were taken at the time the information was given: No. 1 turbine, steam pressure per gage, 175 pounds; superheat, 143 degrees. No. 2 turbine, steam pressure per gage, 175 pounds; superheat, 68 degrees. No. 4 turbine, steam pressure per gage, 175 pounds; superheat, 78 degrees. (No. 1 turbine is nearest the boilers, giving the highest degree of superheat.) Superheated steam has been in use here for about six years and no difficulties have been encountered. The degree of superheat is not believed to have exceeded 150 degrees. Cast-iron fittings are used and cast-iron gate valves with bronze seats and discs have proved entirely satisfactory. No difficulty has been encountered in the use of reciprocating-engine auxiliaries."

Fore River Shipbuilding Company, Quincy, Mass.

"Steam pressure, 120 pounds per gage; superheat, 100 degrees. The superheaters have given some trouble due to pitting of tubes. Reciprocating engines have given trouble on account of cutting of valve seats; also seats of globe valves have cut by wire drawing. Valve bodies, seats and discs are generally of composition, except throttle valves, which have cast-iron bodies."

The Philadelphia Electric Company, Philadelphia, Pa.

Information from Mr. W. H. Norris, Mechanical Engineer.

"Steam pressure, 175 to 190 pounds per gage; superheat, not over 60 degrees F. at engine throttles. Reciprocating engines and turbines are used. Additions made to generating plant in past six years are steam turbines. The use of this

degree of superheat has caused no difficulties of any kind and it is believed to give a gain in efficiency as well as doing away with the trouble caused by wet steam and by the accumulation of water in steam piping."

H. R. Worthington & Company, Harrison, N. J.

"This plant was designed for 150 pounds steam pressure per page and 150 degrees normal superheat. The superheat has been reduced to 100 degrees, and it is found that all parts of the installation work better and the cost of repairs is reduced under these conditions. It is found that there is trouble with valves and valve seats of reciprocating engines when the degree of superheat is above 80 to 100 degrees. There has been considerable trouble with leaking valves. These generally are of composition 88–10–2. No difficulty at all has been experienced with superheaters."

The Babcock & Wilcox Company, Bayonne, N. J.

"Steam pressure, 195 pounds per gage; superheat, about 70 degrees. Commercial high-pressure valves and fittings of cast iron are generally used and cause little trouble. No more than the ordinary overhauling appears to be required for reciprocating engines running under superheated steam."

West Jersey & Seashore Railroad, Westville, N. J.

"Steam pressure, about 175 pounds per gage; superheat, about 125 degrees. It has been found necessary to keep all conditions as nearly uniform as possible. Any priming of the boilers may cause the temperature to drop to that of saturated steam and this is followed by a gradual rise to the temperatures usually maintained. These variations have made it necessary to discard cast iron and semi-steel valves and fittings and they have been replaced with cast-steel fittings; and valves with cast-steel bodies, bronze discs and nickel seats. Superheated steam is also used for driving the auxiliaries with reciprocating engines and has caused an irregular wear in some cases. Cast-iron packing for piston rods and valve stems has been found satisfactory.

Hudson & Manhattan Railroad Company, New York City.

"Steam pressure, 175 pounds per gage; superheat varies with load carried and distance of turbine from the boilers. Upper limit, about 125 degrees; lower, about 30 degrees. Valves and fittings are of cast iron with composition seats for valves. They have given no trouble during the two years that superheaters have been in use."

Chattanooga Railway & Lighting Company, Chattanooga, Tenn.

"Steam pressure, 175 pounds per gage; superheat, 125 degrees. Composition valves were not found satisfactory and have been replaced by valves of Crane's superheat metal, which are giving satisfaction."

Chicago & Milwaukee Electric Railroad, Highwood, Ill.

"It is stated that in this plant there are two boilers which are provided with superheating surface sufficient to give about 8 to 10 degrees superheat. These boilers, with several others, are used to supply steam for two compound reciprocating engines with condensers. The increase in coal economy when the boilers having superheaters are in use is said to be noticeable."

General Electric Company, Schenectady, N. Y.

"Steam pressure, 185 pounds per gage; superheat, 150 degrees to 200 degrees. Several superheater tubes have been burned out. Superheated steam is in use for both turbo-generators and auxiliaries. Cast-iron valves and fittings have given trouble from cracking and getting out of alignment under high temperatures. Cast steel has given the best satisfaction for valves and fittings."

Amoskeag Manufacturing Company, Manchester, N. H.

"Steam superheated at 30 degrees F. is used here and is found to be economical. The superheaters have given no trouble at all. At this temperature the ordinary commercial fittings and valves are used."

Kansas City Metropolitan Street Railway Company, Kansas City, Mo.

"Steam, 175 pounds per gage; superheat, 75 degrees."

Pfister & Vogel Company, Milwaukee, Wis.

"Steam, 160 pounds per gage; superheat, 100 degrees. This company has put a separator (a large steam drum with baffles) in the main steam pipe for the purpose of removing particles of dust which were carried with the superheated steam and which gave trouble by cutting the cylinders and valves."

Denver Tramway Company, Denver, Col.

"Steam pressure, 175 pounds; superheat, 100 degrees."

International Harvester Company, Chicago, Ill.

"Steam pressure, 150 pounds per gage; superheat, 50 degrees."

Allen B. Wrisley Company, Chicago, Ill.

"Steam pressure, 125 pounds per gage; superheat, 125 degrees."

Mandel Brothers, Chicago, Ill.

"Steam pressure, 150 pounds per gage; superheat, 125 degrees."

James S. Kirk Company, Chicago, Ill.

"Steam pressure, 150 pounds per gage; superheat, 150 degrees."

South Side Elevated Railway Company, Chicago, Ill.

"Steam pressure, 165 pounds per gage; superheat, 50 degrees."

Chicago City Pumping Station, Chicago, Ill.

"Steam pressure, 150 pounds per gage; superheat, 75 degrees."

Omaha and Council Bluffs Railway Co., Omaha, Neb.

"Steam, 175 pounds per gage; superheat, 75 degrees."

Twin City Rapid Transit Company, Minneapolis, Minn.

"Steam, 175 pounds per gage; superheat, 75 degrees."

LOCOMOTIVES.

The report of a committee at the American Railway Master Mechanics Convention, published in the "Railway Master Mechanic," August, 1910, states that 805 locomotives on American roads use superheaters. Four hundred and eighty-seven of these belong to the Canadian Pacific Railroad.

SUPERHEATED STEAM AT U. S. NAVY YARDS.

Superheated steam is used in the central power plants of nearly all the U. S. Navy Yards and Stations. The types of boilers most generally used are the Heine and the Babcock & Wilcox, fitted with Foster superheaters. The pressure intended to be carried at the boilers varies from 150 to 200 pounds per gage, and the degree of superheat usually carried is about 100 degrees F. This degree of superheat is, in some cases, regulated by a water spray into the steam pipe where the steam leaves the superheaters. The information which I have collected has been furnished by the commandants and engineer officers of the various navy yards, and from their reports, and from my personal inquiry at several of these stations it is believed that the principal difficulty encountered is due to leaking valves. Composition and cast-iron valves apparently wear away quicker under the action of superheated steam than with saturated steam.

U. S. Navy Yard, Boston, Mass.—The average steam pressure now carried is about 180 pounds per gage at the boilers. One hundred degrees superheat is carried at the boilers, which gives from 60 degrees to 90 degrees at the engine and turbine throttles. No great difficulty has been experienced from the use of superheated steam either in engines or turbines. All auxiliaries, including pumps, are operated by superated steam. The old steam main was taken down in 1909 and was found to be worn quite thin, and this wear was believed to have been caused by superheated steam. The new steam main is of steel pipe with welded-steel flanges and cast-steel valves and fittings.

U. S. Navy Yard, New York, N. Y.—Steam is now carried

on the boilers so as to give 150 pounds at the turbine throttles. The average steam pressure for the past six months at the boilers has been 158 pounds per gage. The average superheat at the same time has been 86 degrees. The boilers have been operated with moderate firing, an effort being made to avoid forcing them. Aside from an occasional leaking tube, the superheaters have given no trouble and no difficulties have been encountered which are directly attributed to the use of superheated steam.

U. S. Navy Yard, Philadelphia, Pa.—At present only four boilers are equipped with superheaters. The power plant is designed for 150 pounds pressure and 100 degrees superheat.

U. S. Navy Yard, Washington, D. C.—The average steam pressure carried at the boilers is 160 pounds per gage. The degree of superheat varies from 108 degrees to 128 degrees. The auxiliaries, including pumps, are run from the main steam line, and no difficulties have been experienced from the use of superheated steam. All fittings and valves are extra heavy and of either cast-iron or semi-steel, and they give very good satisfaction aside from the difficulty found in keeping them tight. The action of superheated steam is found to be very severe on the seats of valves, which require to be made of a special composition in order to get a satisfactory valve. Tauril packing has been found satisfactory for joints.

U. S. Navy Yard, Norfolk, Va.—The average steam pressure carried at the boilers is 155 pounds per gage. The average superheat, 99 degrees. The superheaters have been in use for about two and one-half years. During that time two tubes have been renewed which were evidently of poor quality. Superheated steam is used on all auxiliaries as well as the main turbines and no difficulties have been encountered in its use.

U. S. Navy Yard, Charleston, S. C.—Average pressure carried on the boilers, 150 pounds per gage; average degree of superheat, 98 degrees. Superheated steam is used on the turbines but not on the auxiliaries. No difficulty has been encountered in its use.

U. S. Navy Yard, Pensacola, Fla.—Average steam pressure

carried on the boilers, 150 pounds per gage; average degree of superheat, 10 degrees. The installation is designed to furnish 50 degrees of superheat, but the boilers are run at only about one-third their normal capacity. No difficulties have been encountered in the use of superheated steam. The naval tug *Waban* is also equipped with a superheater.

U. S. Navy Yard, Mare Island, Cal.—The central power plant is designed for a steam pressure of 200 pounds with 100 degrees of superheat. The installation has not yet been completed.

U. S. Navy Yard, Puget Sound, Wash.—Average steam pressure now carried at the boilers, 115 pounds per gage; average degree of superheat, 98 degrees. Superheated steam has been used at this station for about a year. The superheater headers connect with a heavy cast-iron fitting from which steam is taken for all the steam lines, which are from 8 to 10 years old, and have standard cast-iron fittings. No difficulty has been encountered in the use of superheated steam.

U. S. Naval Proving Ground, Indian Head, Md.—In this plant some of the steam mains are from 1,500 to 1,800 feet long and steam is used for drying purposes as well as for furnishing power. Superheaters were at first installed without making any changes in the steam piping or in the engines. This caused very serious difficulties from the start. Pipe joints, valves and valve seats began to leak, and reciprocating engines became inefficient and in need of constant attention. At times it was necessary to force the boilers so that the steam temperatures at the superheaters reached 640 degrees when the normal temperature should have been 585 degrees. The plant was entirely overhauled, the degree of superheat reduced and valves and fittings of cast iron were replaced by others of cast steel, valves having nickel-composition seats. The pressure carried on the boilers at present is 140 pounds per gage and the degree of superheat at the boilers about 110 degrees. The designed superheat is 150 degrees. The steam engines are being replaced by turbo-generators.

COMMENTS ON SUPERHEATED STEAM.

Mr. Fred. N. Bushnell, Engineering Manager of the Stone and Webster Engineering Corporation, Boston, Mass., states as follows:

"In general, in all steam turbine plants which we build or add to, we install steam superheaters in the boilers. Our latest practice has been to use a steam pressure of 175 pounds with a superheat above the temperature of the steam at that pressure and at the rating of the boilers of 125 degrees F. when the boilers are fired with coal. We avoid a steam temperature higher than 550 degrees F. at the boilers under all circumstances. The above temperatures are based on clean boiler tubes, clean superheating surfaces and first-class firing. Dirty water tubes increase the superheat by decreasing the absorption of heat by the water; dirty superheater surfaces decrease the superheat. The higher the furnace temperature the lower will be the superheat, and vice versa, because for a boiler operating at a given capacity there will be less volume of gas with a high-furnace temperature and a more rapid absorption of heat through the water-heating surfaces than with a low-furnace temperature, and consequently the temperature and volume of the gases at the superheater is less For this reason, with oil as fuel and with the same size of boiler and superheater, the superheat is lower than with coal. We use superheaters installed in the boiler setting, and practically the only difficulty we have had with superheated steam when the total maximum temperature was less than 550 degrees F. has been with cast-iron valves and fittings. Valves of semi-steel are also found unsatisfactory. We use cast steel where cast iron was used before. Except the automatic stop valves on the boilers and the throttle valves, we use gate valves entirely with superheated steam, and have found cast-steel discs against nickel-bronze or nickel-steel seats satisfactory. In safety valves we use nickel-bronze, or nickel-steel seats and discs. With the limits of temperature and pressure which I have given we have had no difficulty in keeping valves, packing and joints tight.

"In some plants using reciprocating engines we have installed superheaters giving about 50 degrees F. above the temperature due to the pressure at the boilers. This is sufficient to insure perfectly dry steam at the throttles of the engines. On auxiliary engines, however, it is our practice to use steam of full pressure and superheat as furnished steam turbines and we have found no trouble in doing so. It is probable that by the time it reaches the throttle valves of the auxiliaries the temperature has been considerably modified by radiation. Small cylinders and valve chests have been in use for five years under these conditions and it has not yet been found necessary to rebore them.

"We are as careful in the installation of separators in superheated-steam work as we are in case of the use of saturated steam, as we find that in starting up or on load fluctuations the effects of water are as serious as with saturated steam. While the quantity of water may be slightly less, under conditions of starting and load fluctuation, the temperature range is very much greater."

Mr. F. Sargent, Consulting Engineer, Chicago, Ill., states as follows:

"I have had about twelve years' experience in the installation and operation of superheated steam in large electric-power plants, the earlier plants using reciprocating condensing engines and the later ones using steam turbines. In nearly all of our steam-turbine plants we used superheated steam. The best practice we consider to be about 200 pounds pressure per gage and from 125 to 150 degrees F. of superheat. This I have found to be about the degree of superheat used in what are considered plants of excellent design in the United States, England and France. In Germany the tendency is to superheat to about 210 degrees F. with 200 pounds pressure per gage. We are now designing a new turbine plant for the Commonwealth Edison Plant of Chicago, in which each unit will be the equivalent of 30,000 indicated horsepower. In this plant the maximum boiler pressure will be 250 pounds gage pressure and the superheat 125 degrees F. There is,

however, little if any tendency to increase the steam pressure above 200 pounds per gage. We are doing it in this case, but the gain in energy will be so little that we do not think it will become a general practice. The typical turbine plants with which we are or have been connected which use superheat are as follows: The Commonwealth Edison Company, Chicago, Ill.; The Edison Electric Illuminating Co., Boston Mass.; Norfolk and Portsmouth Traction Co., Norfolk, Va.; Twin City Rapid Transit Co., Minneapolis, Minn.

"The experience of users of superheated steam, generally speaking, is that it is economical and that it pays in connection with steam turbines. It is not only economical from the fuel point of view, but it also materially reduces the tendency of erosion of the turbine blades by the action of water carried by the steam. In general we have had no trouble with either boiler or superheater on account of the use of superheated steam. The only case has been very lately with superheaters at the Quarry Street Station of the Commonwealth Edison Company, which is due to the superheater coils warping and for which we have not yet found a satisfactory explanation. The use of superheated steam involves special construction in valves, piping and fittings from the boiler to the engine. The use of brass or cast iron has been found objectionable and generally they do not stand the work. We specify that all valve bodies, fittings, steam chests and parts which come in contact with superheated steam shall be of cast steel. The valve seats we make of nickel-bronze, 75 per cent. of pure nickel, and the valve stems we make of nickel-steel to prevent corrosion. Even neglecting the special qualities of nickel seatings, on account of the great toughness of this metal, and the methods which can be employed for securing rings of this substance to the valves and conical surfaces, it has the special advantage of having a coefficient of contraction and expansion with temperature almost exactly the same as that of Siemens-Martin cast steel, so that no slackness of the rings occurs and the valves remain steam tight. There are instances in which valves constructed with nickel seatings have been satisfactory

when used with steam temperatures of 750 degrees F. We are also now using nickel-steel containing slightly over 30 per cent. nickel for valve seats and valve stems with good results We have found that satisfactory gate valves for superheated-steam work can be furnished by the Crane Company of Chicago. The Edward Steam Specialty Company of this city also make a specialty of angle valves, globe valves, blow-off valves and small special valves for this work. Some special gate valves designed by Ferranti, the English engineer, have been imported by the Detroit Edison Company and have given excellent results. This valve is extensively used in England. For gaskets, at first we used copper, but that soon corroded and gave out. In the large plants in Chicago, the McKim double-sided copper gasket, with asbestos paper between the copper, is now used, but we have not yet found a very satisfactory gasket. The tendency seems to be to do away with gaskets as far as practicable.

"The effect of superheated steam on cylinders of reciprocating engines used for auxiliaries in our plants has given considerable trouble due to the tendency of castings coming in contact with superheated steam to warp out of shape. This has been overcome to some extent by using cast steel for valves and valve chests. It has also been one reason for reducing the degree of superheat in these plants. The tendency is, however, to do away with all reciprocating auxiliaries and to substitute turbine-driven auxiliaries, *i.e.*, the circulating pump, air pump and water of condensation pump are all on one shaft and operated by a steam turbine which either exhausts in a feed-water heater or into one of the lower stages of the main turbine. Several combinations to attain this result are now being tried both in this country and in Europe. We already drive our boiler-feed pumps by steam turbines, and I believe in the near future there will be no necessity of using reciprocating engines as auxiliaries in any of the turbine plants.

"The first large plant which I recall in this country to use superheated steam was the Atlantic Avenue plant of the Edison Electric Illuminating Company, of Boston. When the

first boilers (representing about 25 per cent. of the present capacity) were installed, the use of superheated steam was practically unknown in this country, but we decided to try superheaters when adding to the plant. The results were found so satisfactory that the remainder of the plant was equipped with superheaters. The boilers and superheaters are all of the Babcock & Wilcox type and the engines are of the McIntosh and Seymour vertical cross-compound condensing type. The steam pressure is about 175 pounds, and the degree of superheat about 50 degrees F. I have never heard of any trouble with the boilers, engines or superheaters due to the use of superheated steam. The plant, however, is practically shut down at present, as the load is being carried by a modern steam-turbine plant.

"In the year 1903, we installed Babcock & Wilcox boilers with superheaters attached in the power house of the South Side Elevated Railroad Company, Chicago. The working pressure was 200 pounds per gage and the degree of superheat was 120 degrees F. The engines were horizontal cross-compound direct-connected Corlis type of about 3,500 H.P., operated at 80 revolutions per minute. On starting up these engines we had considerable trouble with the lubrication of the cylinders, which became scored. We finally reduced the degree of superheat by cutting out tubes to about 50 degrees F., and the engines and superheaters have worked very successfully ever since.

"In 1905 and 1906, we installed more boilers in this plant, using the same pressure and superheat at 50 degrees F. These were used on two vertical cross-compound condensing Corlis engines of about 4,000 H.P. each and 80 revolutions per minute. This installation has given no trouble whatever, either in the engines, boilers or superheaters.

"In 1902, we designed a new power plant for the Twin City Rapid Transit Company, Minneapolis, Minn. Babcock & Wlicox boilers were installed with 200 pounds pressure per gage and 120 degrees F. superheat. There were four engines of the vertical Corlis cross-compound condensing type, each

engine being about 7,500 H.P. at 80 revolutions per minute. The engines in this plant have given a great deal of trouble on account of superheated steam. The valves have cut badly, warped and cracked; steam chests have warped and cracked and cylinders have scored. The engines are now gradually being replaced by steam turbines.

"We have installed some other reciprocating plants with superheat, but the three mentioned are good examples of the difficulties encountered in similar cases. I am unable to state what the economy due to the use of superheated steam is, but in connection with steam-turbine work it is unquestionably very economical from the coal-consumption point of view. Then, again, the erosion of the turbine blades due to the action of water in the steam is much lessened by the use of superheated steam, and its use is economical from this point of view. In the matter of its use for large reciprocating engines we take an entirely different position. From our experience we should feel that superheated steam of a total temperature above 450 degrees F. would be harmful to the engines, and in our work we should hardly care to go over 425 degrees F."

Mr. George A. Orrok, Mechanical Engineer of the New York Edison Company, states as follows:

"Superheated steam is being used in the plants with which I have been connected, at Utica, 8,000 kw.; Helderberg, 2,000 kw.; Williamsburg, 100,000 kw.; Waterside Station, 200,000 kw.; Gold Street, 60,000 kw. (all approximately), and we are at present at work on a new station which will have a capacity of approximately 100,000 kw. There are also a number of other small stations for which we have done the engineering and which are at present operated by other companies. On all of our new work we are using the same standards that we have used in the past, 200 pounds pressure and from 75 to 100 degrees of superheat. This, we believe, gives us better results than we would get by attempting to superheat to a higher degree. In all the above stations we use the standard Babcock & Wilcox boiler, 14 tubes high, 21 sections wide, with 18-foot tubes, and drums 42 inches in diameter. These

boilers have about 6,000 square feet of heating surface and something over 600 square feet of superheating surface. We are able with this amount of superheating surface to get from 75 to 100 degrees F. of superheat without any additional expenditure of coal.

"The superheaters have always been of steel and show no signs of deterioration. The inner surfaces of the tubes become covered with a reddish oxide which is extremely hard and very thin and which apparently does not grow thicker with age. As far as I have been able to learn, other companies have found very little trouble with their boilers or superheaters due to the use of superheat. In a number of cases where imperfect superheater tubes have been used there has been trouble. In our own station, where we have ninety-six of the large boilers, we have not had to replace more than two or three superheater tubes in five years, which shows that little trouble may be expected from this source. There are other stations in which there has been considerable trouble, but my investigation led me to believe that this was due to imperfect tubes.

"Our steam piping is the standard wrought-steel pipe with Vanstone joints made up without gaskets, as a general thing. But when gaskets are required we use a thin steel gasket about 2/100 of an inch thick and carefully annealed. The pipe fittings are of cast steel made from the cast-iron patterns with slight modifications for fillets, etc. Cast-steel valves are used, sometimes with steel seats and sometimes with nickel seats, depending upon the manufacturer. The Crane Company use a nickel-seated valve which has been found satisfactory. Some of the other manufacturers do not insert any special seat, but machine a seat in the body of the valves. This has been found as successful in our practice as having inserted nickel seats in the valves. We have had no difficulties from the cutting of valves or seats or from the deterioration of fittings from the action of superheated steam. While in Europe I found that most users of superheated steam were using cast iron for valves and fittings. I inquired as to the

composition of the iron used in making these valves and fittings and found in every case that the iron was low silicon containing not over 0.6 of one per cent. of silicon. Such iron is not common in the United States, and there would be no economy in its use here, as the cost would be higher than that of cast steel.

"We have found no ill effects from using superheated steam in properly-designed turbines. The trouble with superheat a few years ago was due to the fact that the heat strains set up in the casings and rotors of the turbines were not well understood. With a properly designed turbine no difficulties due to the variation of the degree of superheat have been encountered. I do not consider it good practice to use superheat in reciprocating engines which have not been designed for its use, as the temperature ranges are so considerable that unequal expansion and contraction is liable to cause trouble. Our engines have been designed with the intention of using superheated steam and the results have been found very good indeed. We had some difficulties at first, but found that they were due to the lubricating oil used and not directly due to the use of superheated steam."

Mr. F. G. Clark, Superintendent Power Station, Pennsylvania Tunnel and Terminal Railroad Company, Long Island City, N. Y., states as follows:

"This company operates three power plants, only one of which is equipped to furnish and use superheated steam, to wit: the Long Island Power Station. This plant contains thirty-two boilers, each rated at 524 B.H.P. The boilers are water-tube type, manufactured by the Babcock & Wilcox Company. They were originally equipped with two sets double-tube B. & W. superheaters intended to furnish 200 degrees of superheat at the boiler with normal rated output.

"As just stated, the plant was designed to produce steam superheated to 200 degrees at the boilers to insure at least 175 degrees at the turbine throttles. The boilers are designed to carry 200 pounds steam pressure, but on account of light load and the necessity for better regulation, operation was

started at 175 pounds pressure which has been increased by 5-pound increments, until at the present time the normal is 190 pounds. It is quite probable that the pressure will be brought to 200 pounds within a few months.

"Operation of the plant began in July, 1905, and in October, 1906, one-half of the top row of superheater tubes was removed, reducing the average superheat by approximately 25 per cent.

"During the past year additional equipment has been installed in the plant, and when the design for these additions was under consideration the question of supplying superheated steam to auxiliary apparatus was given attention. On account of economies to be effected by this change and the simplicity of piping arrangements, the chief engineer decided in favor of the use of superheated steam. This necessitated a further reduction in the average superheat of the plant, and therefore the remainder of the top row of superheater tubes has been removed from all boilers resulting in an average superheat of 100 degrees with boilers running at normal rated output and 120 degrees with boilers running at 125 per cent. of rating. A re-design of the furnaces of the boilers and the installation of larger stokers has increased the capacity of the boilers to such an extent that 125 per cent. of rating is the point of greatest efficiency and therefore the point at which the boilers normally will be operated.

"The experience at the Long Island Power Plant indicates that the temperature of superheated steam cannot be kept constant. When high average superheat was generated the variation in temperature was greater than it has been lately with low average superheat. This variation was found to depend principally upon the fluctuating load carried by the plant, the temperature of superheated steam being almost directly proportional to the boiler rating. The temperature varied from 120 degrees at 75 per cent., to 250 degrees at 160 per cent. of boiler rating. There are other causes for variable superheat of which the following are examples:

"With a boiler forced to meet the requirements of a heavy

load, and this load suddenly and greatly decreased, there are intervals of time at which but a small quantity of steam passes through the superheater. At this time the superheat rises from 50 to 200 degrees above normal, and there are records of a number of cases where the total temperature exceeded 750 degrees. This condition has been remedied by the installation of safety valves on the superheater outlets, set sufficiently low to blow off whenever a sudden and large decrease in load may occur.

"The manipulation of feed valves was found to cause a variation of superheat from 25 to 125 degrees. A water-tender usually chokes down the boiler feed during the period of heavy call for steam. This means that a large amount of water must be put in the boiler later on to bring the level to normal position. This produces a drop in temperature of the steam due to a greater proportion of the heat energy being absorbed by the cold water. The subsequent rise is usually to a point above the nomal superheat. These variations can be obviated by the use of feed-water regulators.

"During the first year of operation the high and variable superheat caused some trouble in turbines and valves, but had no effect whatever on piping and very little upon the fittings. We are satisfied that the troubles were due to the frequent and violent fluctuations rather than the high temperature.

"The first change in superheaters, wherein 25 per cent. of the surface was removed, was due to a better understanding of the relative economy. We found that there was practically no net gain in station economy with superheat from 100 to 175 degrees at the turbine throttles, whereas there was a loss with superheat above 175 degrees. The reduction of superheat by 25 per cent. reduced the fluctuations of temperature and had a material effect upon repairs to apparatus subjected to superheat.

"The second change, resulting in the removal of one-half of the original superheater surface, was due to an endeavor to effect greater economies by supplying all auxiliary apparatus

with superheated steam averaging 100 degrees and which should have very little variation.

"We have as yet had but very little experience with superheated steam on auxiliaries, but we have closely followed the operation at such plants as the Port Morris Station of the New York Central & Hudson River Railroad Company, and the Waterside Station No. 2 of the New York Edison Company, where superheated steam is used on auxiliary apparatus with very satisfactory results.

"The troubles due to superheat have all been traced to the changes in cast iron, semi-steel or bronze due to the variable temperature. In the turbines there were distortions of the castings which were very noticeable when high superheat was used, but which cause us no concern whatever at the present time. The semi-steel gate valves and the boiler stop valves have given considerable trouble and have all been replaced by cast-steel valves. There are some large semi-steel fittings in the main line which have shown no indication of growth or distortion, and a number of the smaller size fittings which were originally of semi-steel have been replaced with steel.

"Where superheat of 100 degrees or over is used all valve bodies and fittings and all pipes should be of steel. We have used two or three bronze alloys with success for seats and discs and consider nickel-bronze to be the best of the bronze alloys. We have also used nickel-steel and Monel metal. All the seats and discs of valves lately purchased are equipped with Monel metal. The bodies of some of the smaller valves are also of Monel metal.

"It is our practice to have all flanged joints raised 1/16 inch inside of the bolt holes and given a smooth tool finish. All of this sort of work is very carefully laid out and the flanged joints made with corrugated copper gaskets. We have had practically no trouble where these joints have been properly made.

"The cost of repairs to small valves is somewhat greater with superheated steam than with saturated steam.

"The disadvantages in the use of superheat depend some-

what upon the design or the method of adapting the superheat to plants designed for saturated steam and, aside from the question of economy, can be generally attributed to the effect of varying temperature upon the apparatus."

Mr. W. L. R. Emmet, Engineer Lighting Department, General Electric Company, Schenectady, New York, states as follows:

"A large proportion of all the turbines which we have sold are used with various degrees of superheat, and most of the turbines which we have tested in Schenectady have been tested both with superheated steam and saturated steam, so that the rate of gain in water rate has been determined in many cases with various degrees of superheat. Our total production of turbines of larger sizes has been in connection with electric work on shore, and as a rule the purchasers of our turbines do not adopt steam pressure above 200 pounds per gage. The nature of Curtis turbines is such that we can use much higher steam pressure without practical difficulties and can realize a fair proportion of the increased economy incident to the use of higher pressures. With increased steam pressure we can make a gain in net economy almost proportionate to the increased available energy afforded by higher pressures, and in larger machines no serious practical difficulties would be encountered with pressures as high as 260 pounds per gage. In operating plants using our turbines 200 pounds per gage is about the highest pressure now used, and many users of our turbines have adopted pressures of about 175 pounds per gage. About 200 degrees F. superheat is the highest which I have seen regularly maintained, and one plant in which this degree is carried operates with 175 pounds gage pressure. Many of our turbines are operated with about 150 degrees superheat and pressures as high as 200 pounds. In almost all plants where superheat is used the degree of superheat varies with the load on the boilers, and where the load is subject to fluctuation the steam often runs up to very high temperatures momentarily as a result of the steam flow in superheaters being checked while the firing is heavy. In one plant, where

a pressure of about 190 pounds per gage is used, momentary steam temperatures as high as 700 degrees F. have been reported, although the normal degree of superheat under load is about 150 degrees F.

"The rate of gain in water rate of our turbines with superheat generally amounts to about 1 per cent. for every 12.5 degrees up to 200 degrees. In the case of some machines the rate of gain is somewhat different and in certain cases the rate of gain with high degrees of superheat has been relatively larger than with low degrees. Our experience indicates that a certain degree of superheat gives almost the same percentage of gain in a non-condensing turbine as it does in a condensing turbine, showing that the reduction in the amount of moisture at the low-pressure end, where superheat never exists, is almost exactly equivalent to the difference of superheat. While such large gain in water rate is always shown, my experience has indicated that the gain in fuel economy is very small.

"I find that as a rule superheated steam has no ill effects upon boilers or turbines, or the Curtis type, although there are certain possible exceptions. In the case of the power station of the Norfolk and Portsmouth Traction Company, superheated steam caused a very bad rusting in the pipes, valves and turbines, which was evidently due to some chemical condition in the feed water, and it is quite possible that this corrosion extended to the boilers themselves. I was also once informed by an engineer officer of the British Navy that in certain cases superheated steam had caused very bad corrosion, which was attributed to some chemical action of sea water which was pumped in with the feed. I suppose that the Norfolk case might be something similar.

"In turbines we consider superheated steam more desirable than saturated steam, because it causes much less wear on the blades. Such wear, however, is not a very important factor and might not afford a legitimate reason for using superheat. In one case of a turbine about 12 feet in diameter which had operated with high degrees of superheat it was

discovered, after several years of service, that the casing at the high-pressure end had increased nearly ¼ inch in diameter. It being a vertical shaft machine, this movement had done no particular harm except to put the stationary buckets somewhat out of position. In other cases such distortion of the cast iron by heat might cause serious trouble. In some of our early turbine work we used cast iron for steam chests and with this we had quite serious distortions and cracks developed by superheat. Since we have used steel castings for these parts we have had no such trouble."

The following conclusions are drawn from the reports received:

(a) *Extent of the Use of Superheated Steam in Stationary Plants.*—The principal stationary power plants in this country using superheated steam are furnished with either the Babcock & Wilcox Company's superheater or the Foster superheater, with the superheaters built into the boiler setting. Representatives of both these companies have stated to me that their books show that the sale of superheaters has steadily increased for the past eight or ten years, and that the sales made during the year 1909 were about double those made during the year 1908. The Babcock & Wilcox Company build boilers as well as superheaters, and in 1909 about one-fifth of the stationary boilers which they erected were supplied with superheaters. The list of the users of superheat, which has been given in this article, shows that superheated steam is very extensively used by large corporations who are known to employ the best engineering talent in the design, care and operation of their plants. Their use by many smaller concerns also appears to have been found satisfactory and economical.

(b) *The Limits of Steam Pressure and Degree of Superheat.*—There appears to be a tendency to increase the steam pressure but none to increase the degree of superheat. The concensus of opinion of the most experienced engineers gives about 200 pounds per gage for the steam pressure and 100 degrees to 150 degrees superheat as the best American practice with steam turbines, and not more than 75 degrees super-

heat with large reciprocating engines. In a few cases as high as 200 degrees superheat is successfully used with steam turbines. I have investigated the cases where the use of a higher degree of superheat has been reported, and have found in these cases either that the steam was not used for generating power, or that the higher limit given was due to fluctuations of load.

(c) *Difficulties Experienced with Superheaters.*—The superheaters, which are in use for power plants are nearly all installed in and form part of the boilers. There has been very little trouble with superheaters as usually installed in stationary plants. In general, it can be stated that superheaters of the type referred to give no more trouble than the water-tube boilers in which they are installed. It is quite common to control the degree of superheat by the use of safety valves on the superheater set to blow off at a lower pressure than on the boiler drum, and by sprays in the steam pipe at the superheater.

(d) *The General Experience of Users of Superheat.*—The general experience of users of superheat, within the limits given for the best American practice, viz: 100 to 150 degrees for turbines and not over 75 per cent. for reciprocating engines, is favorable to the installation of superheaters in new plants. Between the limits given above there appears to be no more difficulty in the use of superheated steam than of saturated steam if the plant is properly designed for the use of superheated steam. Where there is liability of variations in load and of priming of boilers, it is difficult to control the degree of superheat. The use of a battery of boilers, part of which are supplied with superheaters and part of which are not, and the use of water sprays to control the degree of superheat, appear to increase the tendency of pipe joints to leak. In many cases where plants were not originally designed for the use of superheated steam the use of a moderate degree of superheat has given excellent results by furnishing dry steam at engine throttles and by doing away with excessive condensation in steam piping and steam cylinders.

(e) *Difficulties Found in the Use of Superheated Steam.*—Few difficulties are found in the use of superheated steam in a plant which has been designed for its use and in which the degree of superheat can be controlled within narrow limits. The use of cast-iron valves and fittings was very general in stationary plants, and it has been found that in many cases the installation of superheaters caused these cast-iron valves to swell, crack or warp, and caused valve seats to leak. On account of this the best practice is to make pipe fittings of cast steel and valve bodies of cast steel with special seats and discs. Nickel-steel valve seats and discs have been found satisfactory as well as seats and discs made of Monel metal and of nickel composition. The subject of the effect of superheated steam on cast iron is very fully discussed in the Proceedings of the American Society of Mechanical Engineers, 1909–1910, and the conclusions drawn from this discussion are given by Professor Ira M. Hollis, as follows:

"The discussion of these papers brings out certain interesting and valuable conclusions which cannot fail to assist in the proper use of cast irons for parts of machinery and boilers. Previously existing differences of experience with this metal under a high temperature are shown to be due to fundamental differences of chemical composition or to variations in the temperature. From this point of view, existing data, even though conflicting, can probably be reconciled.

"The following conclusions as the result of the papers and discussions may be studied with profit in connection with new construction: Cast iron varies in its behavior under high temperature, starting from about 450 degrees F. In many cases it deteriorates in structure and strength to a marked degree. The effect of high temperature is independent of the medium producing it, whether superheated strain, hot gases or solids. The change of structure or deterioration is much increased by a fluctuating temperature. Where the temperature is constant, even though as high as 600 or 700 degrees F., the change in cast iron is not serious enough to prohibit its use, but where the temperature varies considerably, the metal is certain to

develop cracks and distortion that render it unsuitable for steam pipes and other parts under steam pressure.

"Cast iron of certain chemical constituents increases materially in volume when subjected to fluctuating temperatures above 500 degrees F. The chemical composition of the cast iron has a material bearing upon the change of shape and volume and upon the development of imperfections.

"Certain facts in this connection are well shown by Professor Rugan's experiments. As he states, cast iron containing only combined carbon does not change even under high fluctuating temperatures unless the carbon begins to separate into a graphitic form. Free carbon is one of the factors assisting in the deterioration under high temperature, especially when associated with silicon. The latter seems to be the chief cause of increase in volume. Where the free carbon is constant the growth in dimensions is roughly proportional to the percentage of silicon.

The use of cast-iron fittings for superheated steam is inadvisable where the temperature is likely to fluctuate, but it can be safely used where the temperature is to be constant. Cast-iron fittings should not be placed in any parts of a steam-pipe line where there are serious bending stresses in addition to the stresses produced by internal pressure, unless the combined stresses are fully allowed for, or neutralized by expansion joints."

(f) *Economy Due to the Use of Superheated Steam.*—Reports on this subject do not agree, and some very prominent engineers insist that the only economy due to the use of superheated steam is from cutting down the cost of repairs which is made possible by the use of dry steam. Several reports where direct-fired superheaters were used show a loss instead of a gain in coal economy. The following quotation from a letter written by Mr. I. E. Moultrop, Mechanical Engineer of the Edison Electric Illuminating Company, Boston, Mass., expresses the views of many engineers interested in large power plants in which the superheaters are installed in the boiler setting:

"With a steam pressure of 175 to 200 pounds per gage I expect a saving of about 1 per cent. in steam consumption for each 12 per cent. of superheat with steam turbines of the Curtis type varying in size from 5,000 to 12,000 kw. and run with a back pressure of about 2 inches absolute. My personal opinion is that it costs very little to obtain 150 degrees superheat. If superheated steam is to be used the plant should be designed with that end in view. The steam piping should be short and the packing, valves and fittings must be selected with a view of their use with superheated steam."

Many reports of tests show a considerable increase of coal economy due to the use of superheaters. As a rule these trials have been of short duration and have been made by competent experts after careful preparations. They are not entirely satisfactory for the conditions existing in common practice. However, the considerable increase in the installation of superheaters during the past year may be considered evidence that the belief that the use of superheated steam is economical and safe is becoming more general.

If exhaustive trials were made to accurately determine the economy of superheat, the highest degree of superheat justified for gain in economy could be fixed for plants of design similar to the one with which the trial was made. A very satisfactory trial of this character was made several years ago, by the Rhode Island Company, Providence, R. I., at its Manchester Street Power Station. The results of this trial are given as follows:

	Saturated steam.	Superheated steam.
Date	May 15 to 17	May 22 to 24
Total coal, pounds	443,859	391,000
ash, pounds	41,280	30,529
combustible, pounds	402,579	360,471
water, pounds	4,186,825.5	3,666,460
load, in kw	178,000	180,700
Pounds of water per pound of coal	9.435	9.377
kw.h	23.521	20.29
pound of combustible	10.4	10.17
coal per kw.h	2.493	2.163
combustible per kw.h	2.261	1.994

	Saturated steam.	Superheated steam.
Average temperature—		
water entering heater, degrees F.....	61.8	61.22
leaving heater, degrees F.....	151.28	143.98
economizer, degrees F.....	231.39	232.43
gases entering economizer, degrees F.....	543.1	514.2
in main flue, degrees F.....	339.47	319.67
Average draft.....	.808	.473
pressure at boilers, pounds.....	129.84	137.14
boiler pressure at engines, pounds....	127.37	136.69
temperature of steam at boilers, °F...	361.16	462.69
Temperature steam at engine throttle, °F.....	354.6	460.37
Average temperature of steam at receiver, °F..	244.54	235.76
pressure at receiver, pounds.....	11.89	7.79
Average temperature of exhaust steam from condensers entering heaters, degrees F.....	210.39	222.26
Average temperature of water of condensation from exhaust steam from condensers.....	121.15	96.66
Average vacuum.....	25.25	26.04

SUPERHEATED STEAM IN THE MERCHANT MARINE.

Only three vessels belonging to the port of New York use superheated steam. One of these is the steam yacht *Idalia,* and a report of a test made on this vessel to determine the effect of different degrees of superheat on the economy of a marine engine is given in the JOURNAL OF THE SOCIETY OF NAVAL ENGINEERS, Volume XXI, No. 4. These trials were not long enough to furnish data of value as regards actual coal consumption, but they show a saving in steam per indicated horsepower, and indicate a considerable increase in economy of coal consumption.

The boilers of the S. S. *Brazos* of the Mallory Line were fitted with Foster superheaters in 1908. These boilers are of the ordinary Scotch marine type, and the installation is stated to give a marked increase in coal economy and to have been entirely satisfactory to date. Designed superheat about 75 degrees with 210 pounds pressure at boilers.

The S. S. *Creole* of the Southern Pacific Line has Babcock & Wilcox boilers with superheaters. Steam of 210 pounds pressure per gage and about 40 degrees to 50 degrees superheat is ordinarily carried. This vessel has twin-screw reciprocating engines, the cylinders being without steam jackets. Her

record for coal consumption has been excellent as compared with the records of two other similar vessels of the same line for round trips to New Orleans, made at the same season of the year and, therefore, under similar conditions as regards weather. These vessels are the S. S. *Momus* and S. S. *Antilles*. Their boilers are of the Scotch marine type without superheaters. The cylinders are not jacketted. The principal dimensions of these three ships are given in the following table:

SOUTHERN PACIFIC LINE STEAMSHIPS.

	Momus and *Antilles*.	*Creole*.
Length, feet	410	407
Beam, feet	53	53
Draught, feet	25.6	26.7
Tonnage, gross	6,878	6,754
net	4,326	4,302
Cylinders, inches, stroke	34×57×104×63	27½×46½×79×42
Indicated horsepower	7,500	7,000

These vessels all travel between New York and New Orleans on a schedule of five days between ports. The coal consumption, which is given in the following table, is for the round trip in each case, and also includes the coal used in port. The coal consumption given for the *Creole* is for five round trips made in the summer of 1910, while for the *Momus* and the *Antilles* the coal consumption is the least reported for any trips made during the summers of 1908, 1909 and 1910. The total coal consumption and the mean coal consumption is given in the following table:

	Creole.	*Momus*.	*Antilles*.
Tons	1,183	1,412	1,360
	1,150	1,367	1,365
	1,149	1,376	1,256
	1,123	1,445	1,363
	1,139	1,459	1,333
Total for five trips, tons	5,744	7,059	6,680
Mean, tons	1,149	1,412	1,336

During the month of October, 1910, each of these vessels made two round trips, the coal consumption being as follows:

	Creole.	*Momus.*	*Antilles.*
Tons	1,152	1,347	1,339
	1,260	1,573	1,584
Total for two trips, tons	2,412	2,920	2,923
Mean, tons	1,206	1,460	1,462

In 1907 the U. S. Revenue Cutter *Bear* was equipped with a Babcock & Wilcox boiler and a superheater. The ordinary working pressure is about 95 pounds and the degree of superheat from 50 to 75 degrees at the boilers. It was found necessary to install a piston valve on the high-pressure cylinder of the engine as the seat of the slide valve soon wore away. This vessel has been employed in the Behring Sea patrol fleet, and the increase in coal economy since the superheater was installed has been considerable, making the vessel much more efficient for this duty.

The statement is authorized that the installation of direct-fired superheaters on one of the trans-Atlantic liners now being built, is seriously considered. In this case the builders believe that direct-fired superheaters have advantages making them better adapted for marine work. For the same power, the weight, and space occupied are about the same as for boilers without superheaters. Also the superheaters are better located than for boilers with built-in superheaters.

SUPERHEATED STEAM ON VESSELS OF THE U. S. NAVY.

All the vessels in the United States Navy which use superheated steam have Babcock & Wilcox boilers and superheaters. They are as follows: The battleships *Indiana, Massachusetts, Michigan, South Carolina, Delaware, North Dakota;* armored cruiser *New York;* colliers *Vestal* and *Prometheus.*

The U. S. S. *Indiana* was the first vessel to be equipped with superheaters, and these were furnished with the Babcock & Wilcox boilers, which were installed in 1905. This vessel has been in commission for fully two years since repairs were completed, and her chief engineer during this period states that no difficulties at all were encountered which could be attributed to the use of superheated steam. The working

pressure carried varied from 150 to 200 pounds per gage and the superheat was about 40 degrees. Ordinary standard high-pressure composition valves were used and gave no difficulty. Trials were made which gave an increase in economy of about 5 per cent. due to the use of superheated steam.

The following data is taken from reports of the official trials of the vessels named:

U. S. S. *Michigan.*—Average steam pressure at high-pressure steam chest, 246 pounds per gage; average superheat, 86 degrees.

U. S. S. *South Carolina.*—Average steam pressure at engines, 245 pounds per gage; average superheat at engines, 47.5 per cent.

U. S. S. *Delaware.*—Average steam pressure at engines, 285 pounds per gage; average superheat at engines, 62 degrees.

U. S. S. *North Dakota.*—Average steam pressure at turbine chest, 245 pounds; average superheat at turbine chest, 60 degrees. The *North Dakota* has Curtiss turbines. The other three vessels have reciprocating engines.

These four vessels have not been in commission long, and trials to determine the economy due to the use of superheated steam have not been made. Valves coming in contact with superheated steam have cast-steel bodies and discs, with Monel metal seats. The valve seats have been found satisfactory, but nearly all the valve discs have been replaced by new ones of Muntz metal. The following quotation is taken from a report of the officer who inspected the valves for the *Michigan* and *Delaware:*

"The reducing valves for the *Michigan* and *Delaware* failed to stand the specified boiler pressure of 295 pounds with steam superheated about 100 degrees. When tested at the shipbuilding company's works with a total steam pressure of about 475 pounds, the composition piston and liner stuck fast to each other under the superheat. The pistons and liners were changed to Monel metal and no more trouble was experienced.

"Steel springs sometimes take a set and refuse to operate

properly when superheated steam is used. Some trouble has also been experienced with packing of valves. It is also hard to keep valve seats tight with superheated steam unless a special metal is used. Monel metal has given the best results for this purpose."

The U. S. S. *New York* has been in commission about eighteen months. Average steam pressure, 162 pounds per gage at the boilers; average superheat at the main engines, 35 degrees. There has been no difficulty with the superheaters, and none with the main engines which is attributed to the use of superheated steam. The Blake air pumps have the old type enclosed-valve operating gear, and last summer gave considerable trouble from stopping, and upon examination the lever would be found bent and bushings frozen on the pins. The lever was made stronger, bearing surfaces on pins increased, and cups for spraying water in the cylinders as required were fitted. Since this work was done the pumps have worked satisfactorily. No lubricant was used in the cylinders, and it is likely that the trouble with these pumps was caused by the use of superheated steam.

The colliers *Prometheus* and *Vestal* have merchant crews, and the reports regarding the machinery of these vessels have not been entirely satisfactory. In September, 1910, I made an examination of the machinery and boilers of the latter vessel and attributed the loss of life of the high-pressure piston packing rings and the cutting of valve seats of dynamo engines to the use of steam of too high a degree of superheat. A total temperature of steam above 500 degrees was reported to have been carried at times. This report was probably correct, as on several short trips the vessel steamed at about trial speed with three-fourths boiler power. The boilers of these two vessels are not of the ordinary naval type, as the tubes are all 4-inch outside diameter. The main cylinders and high-pressure valves were also examined and were found in excellent condition.

All superheaters of the United States naval vessels are arranged so that the superheaters can be cut out. Reports from

the chief engineers of these vessels state that they are always kept in use. No difficulties attributed to the use of superheated steam have been reported, other than those mentioned above. In general, the reports from vessels of the United States Navy which are equipped with superheaters are favorable to the use of a moderate degree of superheat, with new installations of piping and auxiliary machinery. The use of superheated steam in port should decrease the coal consumption considerably.

ELECTRIC PROPULSION OF NAVAL VESSELS.

By W. L. R. Emmet.

Propositions were recently submitted by the General Electric Company under the writer's direction to the Government for propelling machinery for one of the new battleships, and to various shipbuilders for propelling machinery for one of the Government colliers recently authorized by Congress. The following descriptions and explanations concerning battleship equipment are practically the same as those which have been submitted to the Department with the proposition, it being desirable that the engineering features of the propositions actually considered in these cases be discussed on their merits.

Such a detailed description of an extensive installation is likely to give the impression of complication, but a careful consideration of these designs will show that the arrangement is extremely simple, involving nothing which does not constitute common practice in other industries, and being for a number of reasons simpler and essentially more dependable than similar electric-power applications on shore. The most important of these simplifications arises from the fact that on shore electric power is always developed for a variety of purposes and is delivered to users at a constant voltage and a constant frequency. If the system of distribution from which power is obtained is a large one, the supply of electric energy is practically unlimited and the destructive effects of short circuits and other accidents are apt to be serious. In a ship the electricity would be generated simply for the purpose which it has to accomplish, and the ultimate capacity of the generating apparatus would not exceed the safe carrying capacity of any conductor used in the system.

Another great simplification and advantage is that the fre-

quency need not be kept constant, so that speed changes are accomplished by regulating the steam supply through the prime mover instead of through the applications of electrical control.

With the battleship proposition submitted to the Government drawings of all parts of the proposed installation were given, but one of the drawings (Plate I) which shows the possible arrangement of this apparatus in the ship will sufficiently illustrate the matters referred to in this paper. By comparing this drawing with the design made by the Department for the installation of Curtis turbines it is seen that the shaft positions in both are the same and that no change in the construction of the ship is necessary for the installation of the apparatus here described. It will also be seen that crowding in the engine-room space is materially less with the electric installation, the leads of piping all being short, and every part more accessible. This drawing includes every piece of auxiliary apparatus provided for in the Department's design and is therefore directly comparable.

APPARATUS PROPOSED FOR BATTLESHIP NO. 35. (U. S. S. "TEXAS.")

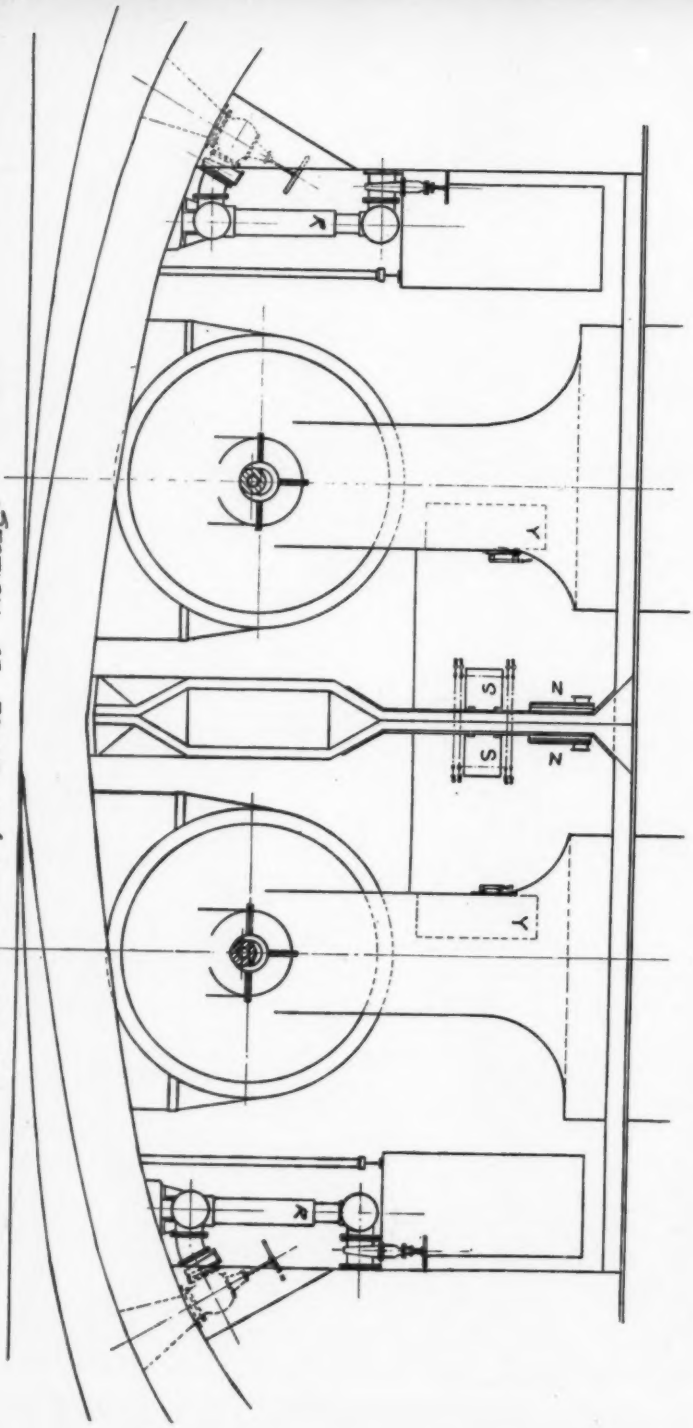
The apparatus proposed for this installation is designed to be placed in the engine-room space shown by the Department's designs for Curtis turbines. It is also designed to drive two propeller shafts situated as shown by the Department's designs, so that the vessel can be built as for the installation of Curtis turbines and can be equipped with the apparatus proposed without appreciable change.

In each engine room it is proposed to install one turbine-driven generating unit, two induction motors, suitable instruments and switching devices, and a group of water-cooled resistances for use in reversal.

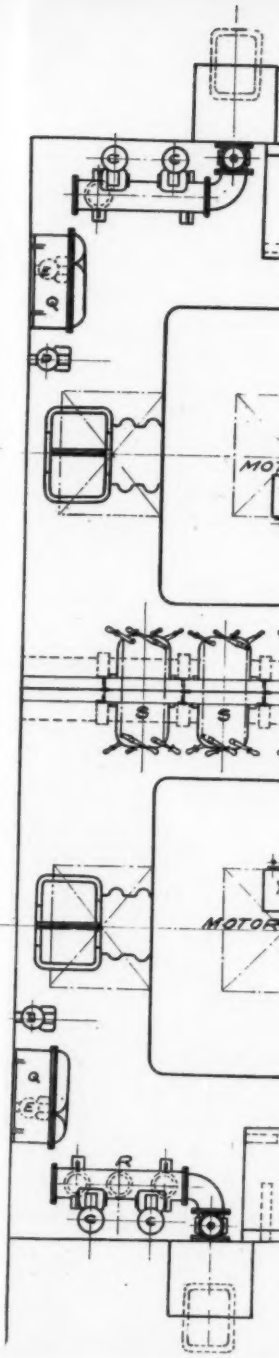
Generating Units.—These units are designed for a capacity of about 12,000 kilowatts, with ample provision for possible overload. The turbine has six two-bucket stages

of the Curtis type, is similar in design to many machines now in commercial use, and is representative of the highest development in steam-turbine production. Its construction is very strong and simple, it operates with large clearances, and its economy is entirely independent of adjustment. The generators proposed are 4-pole alternators, designed to operate at a maximum potential of about 2,300 volts. Their windings are made up of copper bars and are of the simplest and most reliable standard construction. Insulation troubles with such windings are almost unknown, even under the most extreme conditions of heat and moisture. These generating units are to operate at a maximum speed of about 1,250 r.p.m., the frequency under such conditions being about 42 cycles per second. The turbines are equipped with a group of admission valves of standard construction which are under the control of a speed governor. The controlling mechanism is also actuated by a hand lever, so arranged that it will restrict the opening of valves to any desired number or will hold them all closed when necessary. The governor limits the speed when the load is disconnected and this affords a safety limit. In addition to this admission mechanism, an automatic release throttle valve of the Schutte & Koerting type is furnished with this unit. The release mechanism of this valve is connected to an entirely independent speed-actuated tripping device so that excess speed is doubly guarded against. In considering this matter of excess speed it may be well to mention that the construction of both the generator and turbine is such that excess speed cannot do damage outside of the apparatus itself, so that there could be no possible danger to vessel in case of a runaway.

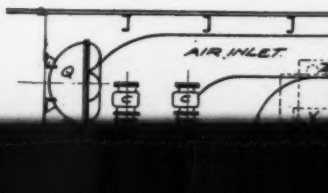
Motors.—To each propeller shaft two induction motors are coupled, these motors being connected together so as to form a compact unit and being so arranged that they can be disconnected and removed separately. One of each pair of motors has a rotor of the squirrel-cage type, and this motor is so arranged that its stator can be connected either for 30 or 50 poles, the 30-pole connection being for high-speed

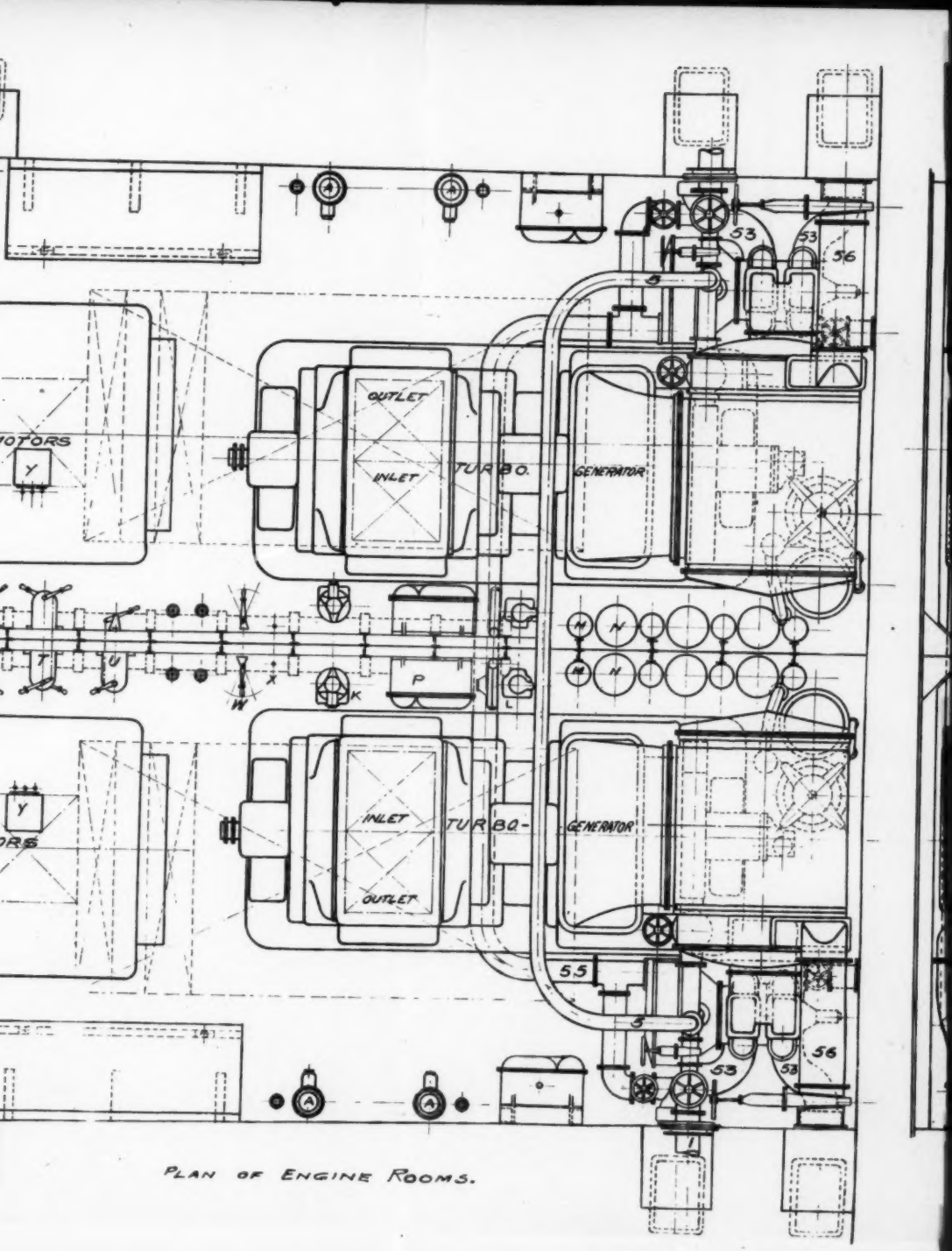


SECTION AT FRAME 104 LOOKING FORWARD.

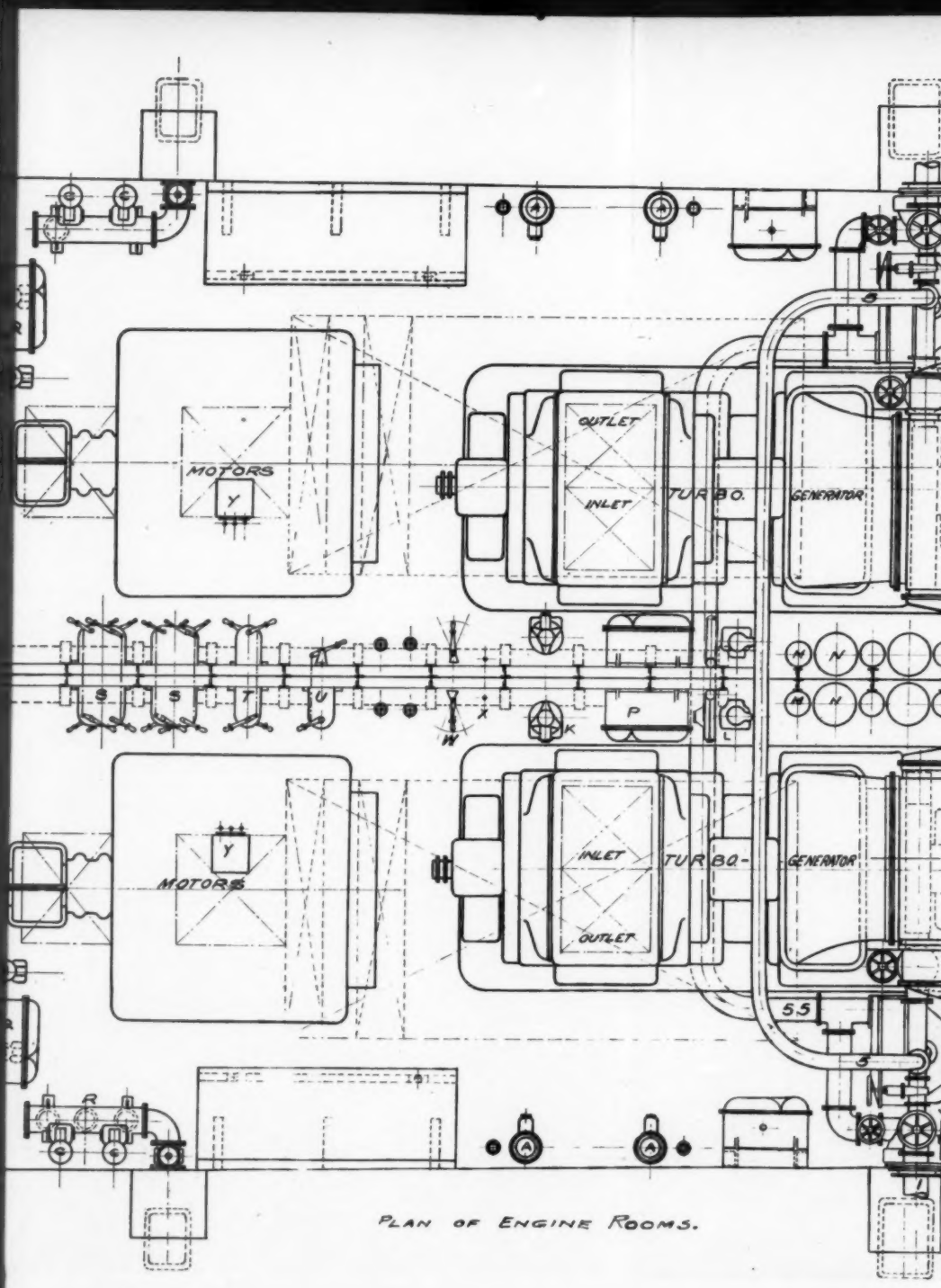


PIPING LIST.		
No	Size	PURPOSE.
1	14"	Main Steam from Boilers.
2	10"	Steam to Turbines

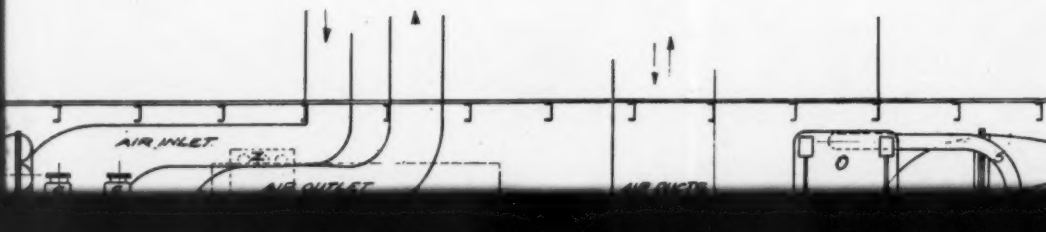


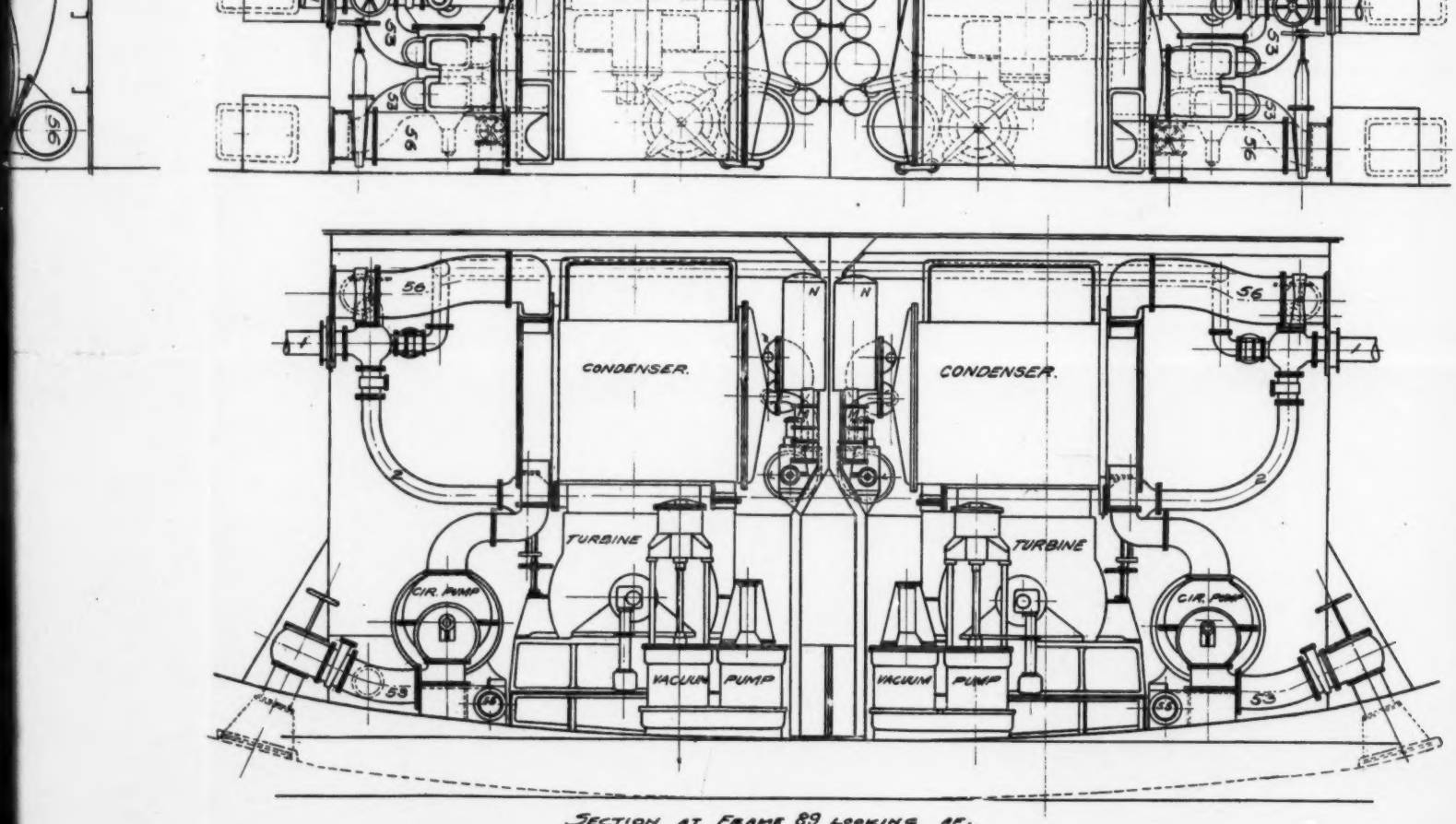


PLAN OF ENGINE ROOMS.



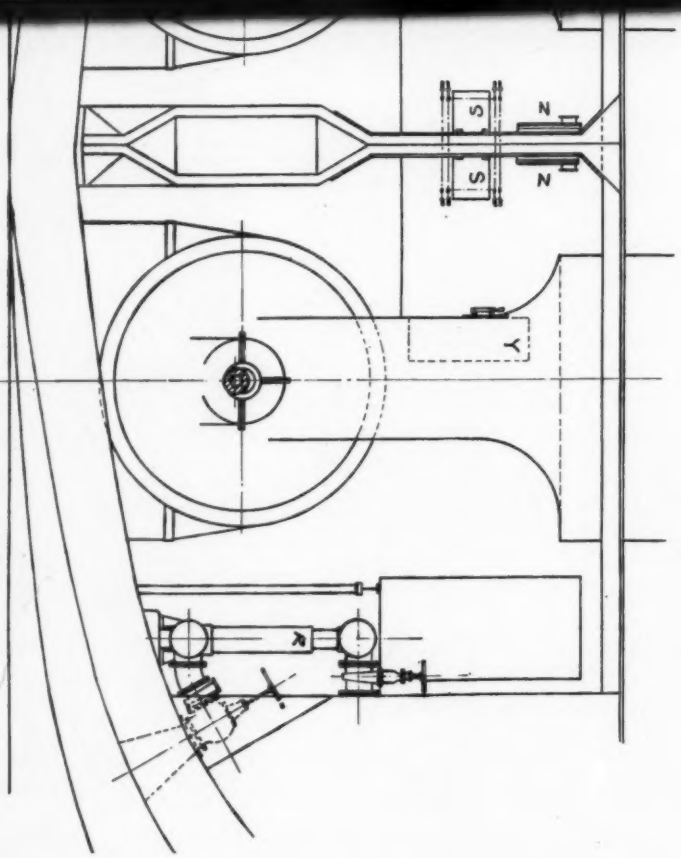
PLAN OF ENGINE ROOMS.

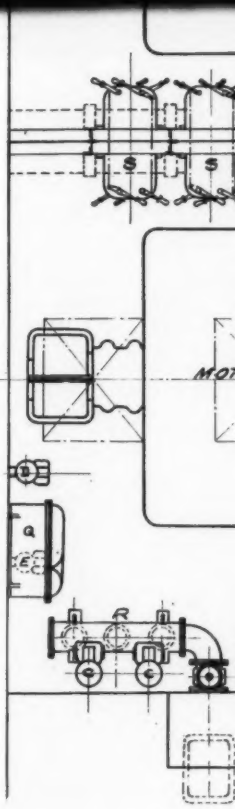




SECTION AT FRAME 89 LOOKING AF..

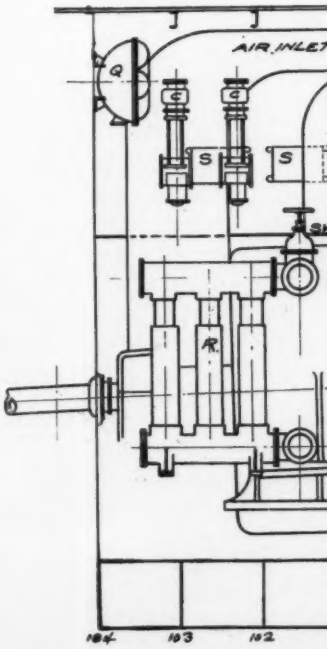
N AT FRAME 104 LOOKING FORWARD.

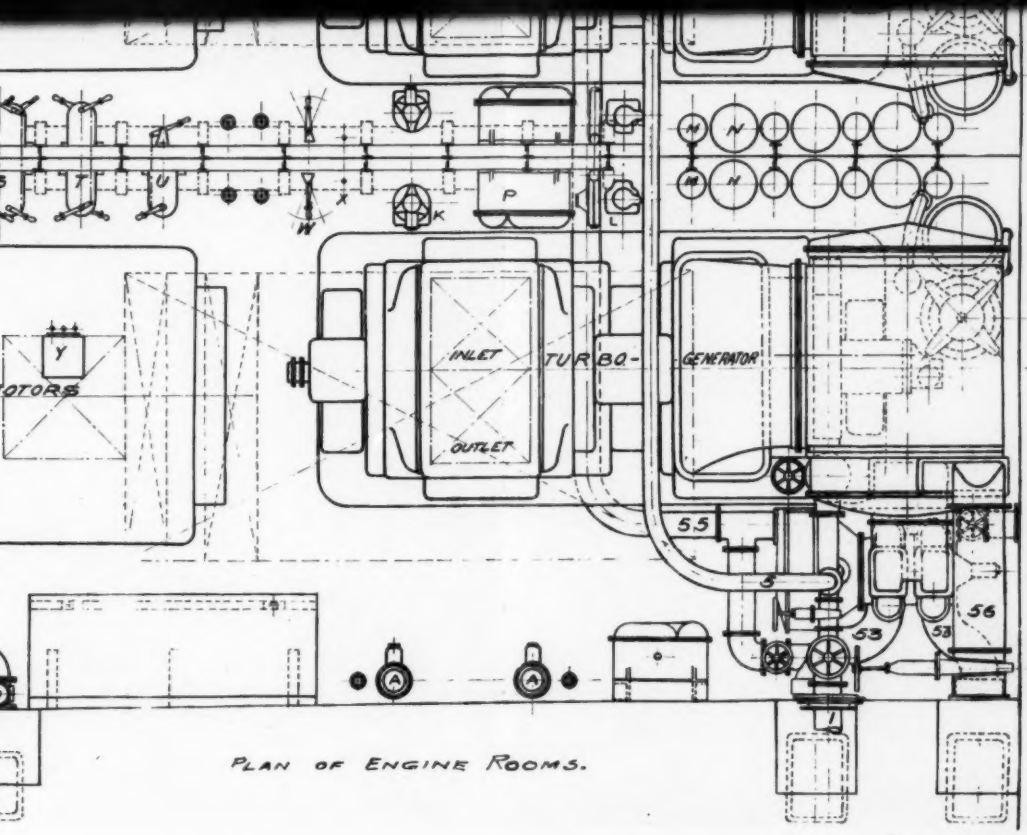




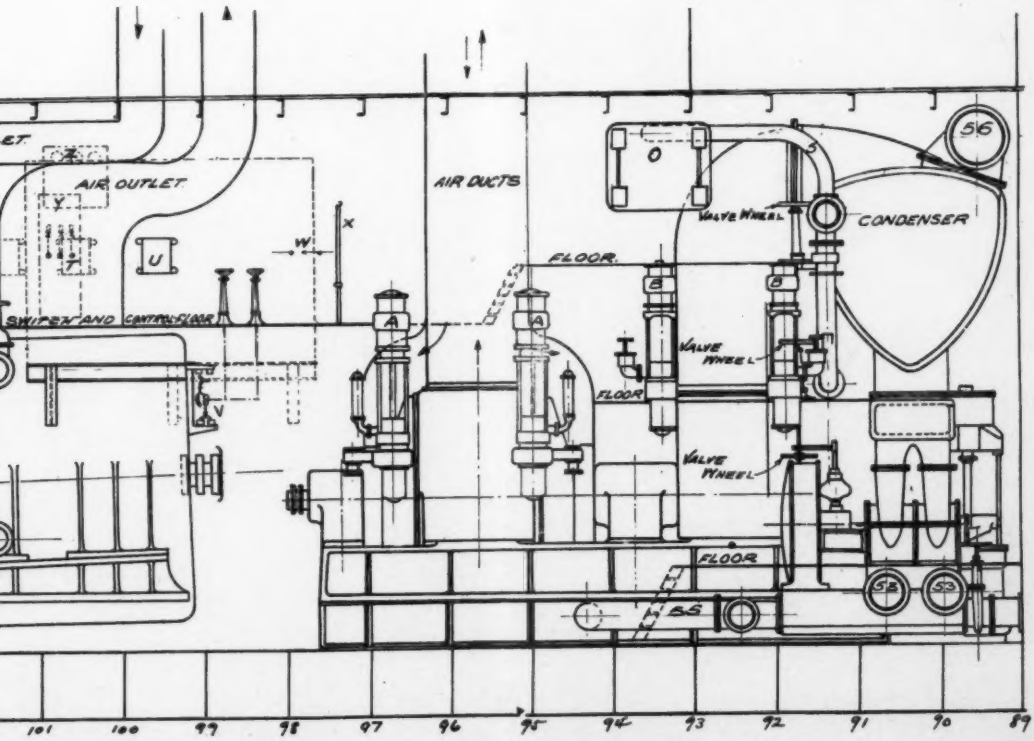
PIPING LIST.		
No	Size	PURPOSE.
1	14½"	Main Steam from Boilers.
2	10"	Steam to Turbines
3	9"	Connection between Main Steam in Port and Starboard Engine Rooms.
53	22"	Main Circulating Pump Suction from Sea.
55		Main Circulating Pump Suction from Main Drain.
56	30"	Main Circulating Pump Discharge overboard.

AUXILIARIES.	
A- Main Feed Pumps.	15½ x 9½ x 24
B- Fire and Bilge Pumps.	12 x 10 x 18
C- Forced Lubrication Service Pumps.	7 x 8 x 12
D- Fuel Oil Pumps.	10 x 9 x 12
E- Oil Cooler Circulating Pumps.	7 x 8 x 12
F- Pipe Insulator Circulating Pumps.	6 x 8 x 12
K- Auxiliary Air Pumps.	7½ x 14 x 12
L- Auxiliary Circulating Pumps.	5 x 6 Engine
M- Air Compressors.	
N- Compressed Air Tanks.	
O- Feed Heaters.	
P- Auxiliary Condensers.	
Q- Oil Coolers.	
R- Water Rheostat.	
S- Motor Switches.	
T- Generator Switches.	
U- T.I.S. Switch.	
V- Pole Changing Switches.	
W- Hydraulic Gear Operating Wheels.	
X- Liquid Tachometer.	
Y- Field Rheostat.	
Z- Switch Panel.	

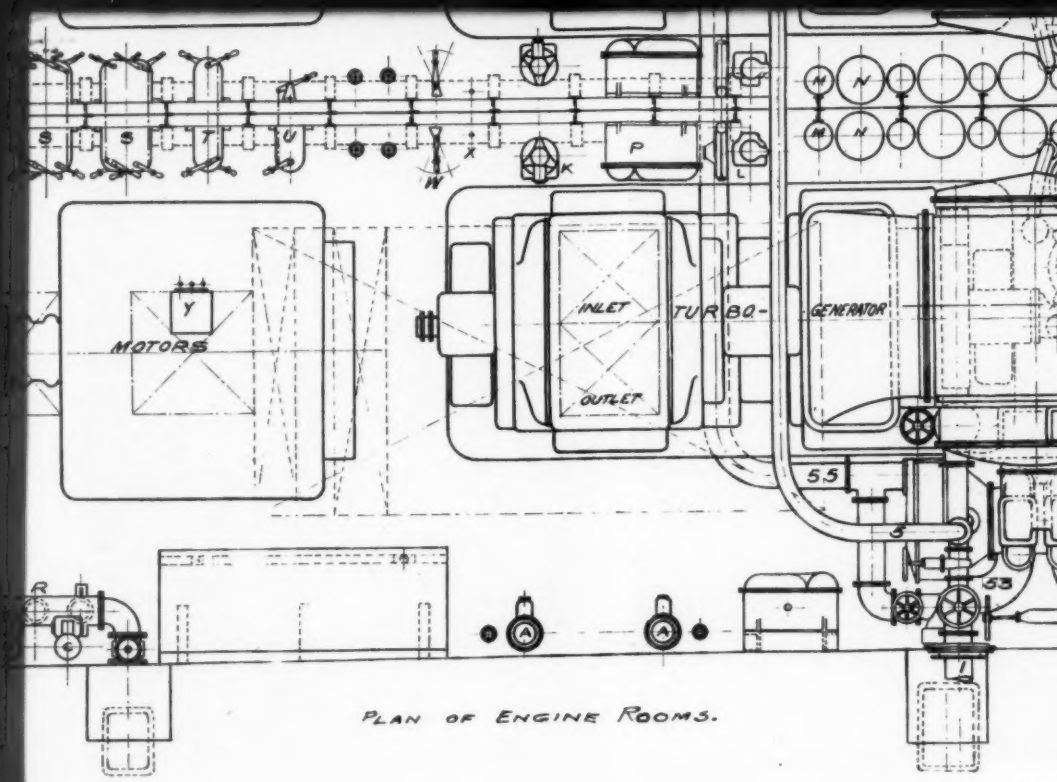




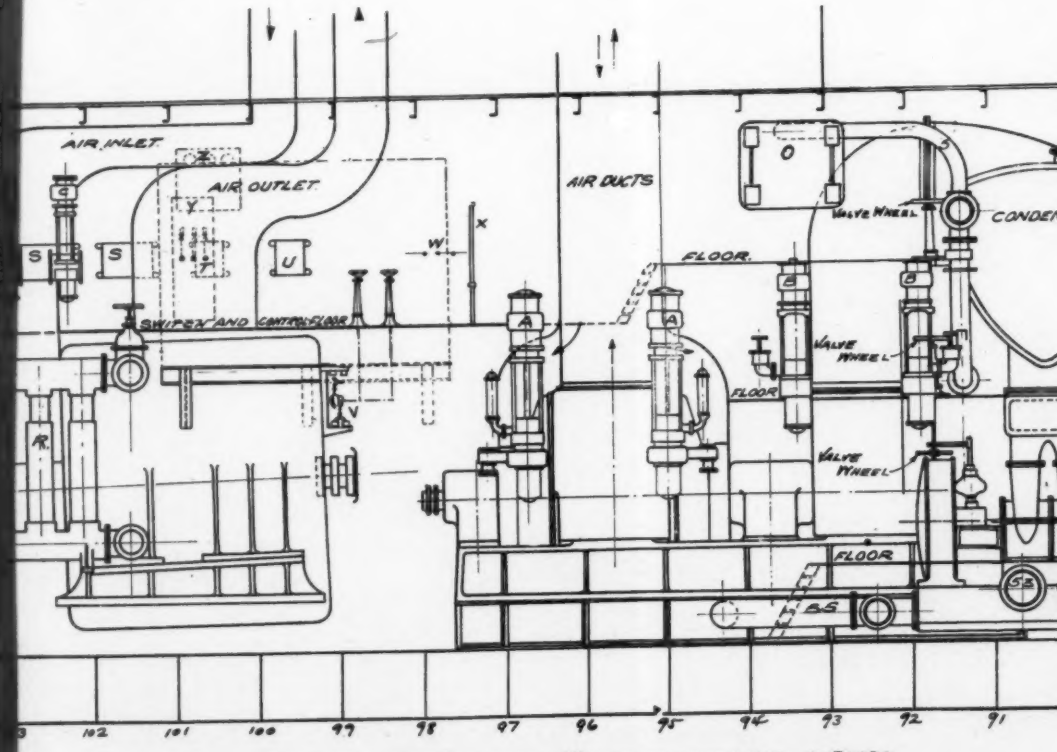
PLAN OF ENGINE ROOMS.



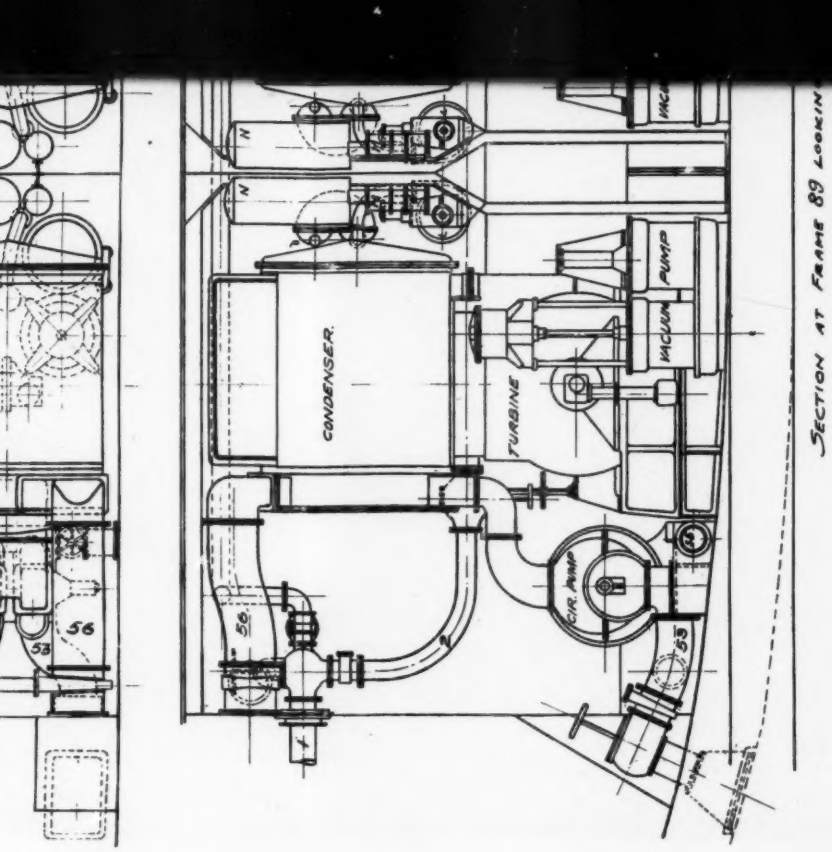
ELEVATION OF STARBOARD MACHINERY LOOKING INBOARD.



PLAN OF ENGINE ROOMS.



ELEVATION OF STARBOARD MACHINERY LOOKING INBOARD.



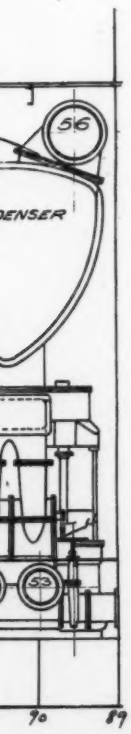


Plate I.

PROPOSED ARRANGEMENT OF ELECTRIC PROPULSION MACHINERY FOR 28,000–TON BATTLESHIP NO. 35, THE U. S. S. "TEXAS."

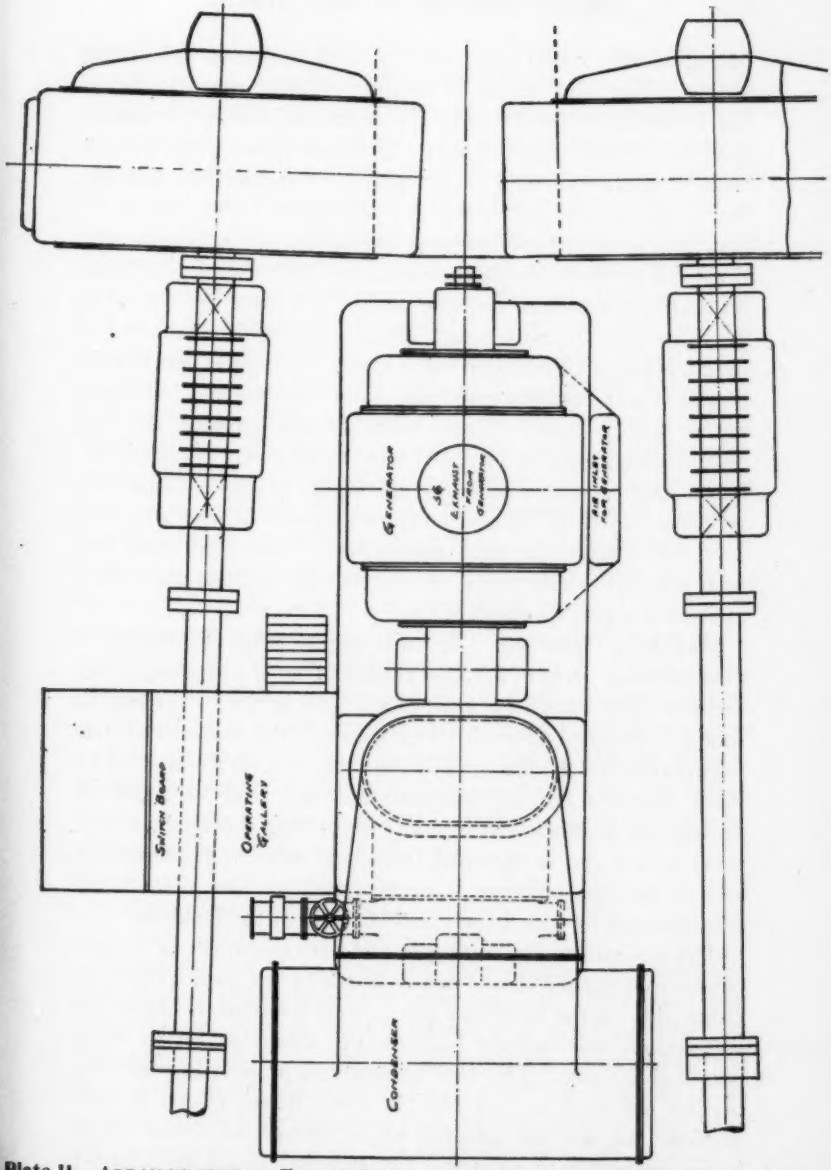


Plate II.—ARRANGEMENT OF ENGINE ROOM FOR ELECTRIC PROPULSION OF A U. S. S. COLLIER.

operation and the 50-pole connection for operation at cruising speeds. The other motor is designed without the pole-changing connections for 30 poles, is suited to operate in parallel with the 30-pole connection of the former motor, and is built with a definite wound rotor suited to connection through slip rings to the water-cooled resistances. This motor is equipped with a lever-operated mechanism by which the slip rings can be short circuited so that the rotor winding is closed on itself or opened so that the currents of rotor are forced to pass through the resistances. The pole-changing of the motor first mentioned is effected by a group of toggle switches of large current capacity and very solid construction, mounted on the motor itself. These switches are actuated through mica-insulted toggles, so constructed as to give a perfect insulation and ample mechanical strength. These connecting switches are of a highly developed standard type which is perfectly reliable for such apparatus. Their construction is such that the whole group of switches can be thrown with a moderate degree of effort.

Switching Apparatus.—In each engine room is installed a disconnecting switch for the generator and a disconnecting and reversing switch for each motor, and in one of the engine rooms is installed a switch which connects the bus-bars of that engine room to the bus-bars of the other engine room. All of these switches are so designed that they can be mounted against the bulkhead with shafts extending through, so that every switch can be operated from both sides of the bulkhead and so that the position of every switch on one side of the bulkhead can be seen from the other side of the bulkhead.

The mechanism for cutting in and out resistances of motors is also so arranged that it can be operated for both motors on either side of the bulkhead. It is thus possible to handle all connections and operate both motors ahead or astern from one engine room. The type of switches proposed operate by toggles and give a very reliable and efficient electrical connection, but are not adapted to be opened or closed while heavy currents are passing. Oil switches might be adopted

which could be opened under load, but those proposed are thought to be simpler and better for the purpose. These switches are arranged with electric locks which are actuated by the exciting circuit so that none of them can be moved while the field of generator is energized. As soon as the field switch is opened all the switches are free to move and can be put into the desired positions. Similar locking devices are applied to mechanism for cutting in and out motor resistances and also to the pole-changing arrangements of motors. There is also a simple mechanical interlock between the two switches which operate motors on the same shaft, which makes it impossible to connect them for opposite directions of rotation.

Resistances.—The resistances for use with motors are made up of coils of non-corrosive metal arranged in vertical wooden ducts, as shown by the drawing. These resistances are kept cool by convection of water, and the cast-iron headers at their top or bottom can either be connected to the circulating system or to separate sea cocks so that they get water from outboard. In the layout of installation submitted herewith the latter arrangement is adopted. These resistances have been developed by careful experimenting, and with a proper water supply are capable of dissipating the total electrical energy which can be generated by the plant. They are designed with very ample margin and, if desirable, could easily be given a greatly increased capacity. Each resistance unit is to be equipped with a large removable strainer so that solid matter cannot get to its interior.

OPERATION.

When the ship is operated at higher speeds both generating units and all four motors will be in use. All motors will then have the 30-pole connection, affording a speed reduction in the ratio of 7.5 : 1. Under such condition of operation the generators could either be run in parallel or separately. It would probably be simplest at such times to operate the engine rooms entirely separately, but if it was desirable to exactly

equalize propeller speeds to prevent racing in a seaway or for any other cause, it could be done by paralleling the generators. When both generators are running the two propellers can be stopped, started or reversed independently, and at such times it might be desirable to have a complete operating force in both engine rooms, although all the changes of connections could be handled from either one. Arrangements might easily be made by which steam admission and pole-changing in both engine rooms could be operated from either one, but it seems unnecessary to make such provisions. Variations of speed from the maximum down to about 15 knots would be effected by a simple change in the steam admission to the separate generating units, exactly as in the case of a ship operated directly by turbines. At about 16 knots the load would become sufficiently light to be carried by one generator and two motors and for all lower speeds the use of one generator can be discontinued and all necessary operations can be handled from the one engine room in which a generator is operated. Even when the ship is operating with high-speed arrangements she can be slowed down to any desired extent by simply shutting off steam. For the sake of economy, however, it is desirable to cut out a generator and motors when they are no longer required.

When it is desired to operated at speeds below 14½ knots it will be more economical to use the 50-pole connection on motors. This condition can be brought about by interrupting the exciting circuit, changing the poles on both motors, and then proceeding to operate from one engine room as before.

To reverse the vessel with powerful torque when either one or two generators are being used it would be necessary to introduce resistance into the motors arranged for reversing and to disconnect the other motors. This is to be accomplished by first opening the field circuit, then moving the switches and resistance devices into the proper positions, and then reestablishing the field circuit. When both the generators are in operation these processes will be gone through with independently in the two engine rooms. When one

generator is in use all operations can better be handled from that engine room. The levers required to move these mechanisms can be easily handled, and experience with such matters shows that the knowledge required for manipulation will be very easily acquired. The design of apparatus is such that the maximum currents which can pass under any combination of circumstances are not sufficient to do damage to any part of the system, consequently wrong connections can simply result in delay and cannot cause burn-outs. The existence of a wrong connection will be immediately apparent from the behavior of the vessel and of the indicating instruments.

When the ship has to be maneuvered in harbors or in narrow waters it may be convenient to keep the resistance in motors continually connected. The motors, in spite of the energy delivered to the resistance, will propel the ship with fair economy under such conditions and the process of sudden reversal will be made more convenient, although reversal under any conditions of operation can be accomplished in a very few seconds.

ECONOMY.

The following tabulation and accompanying curve sheet shows the results which can be accomplished with this equipment at different speeds with corresponding speeds of turbines and propellers. The estimate of power required at different speeds has been arrived at from the best source of information available, and the apparatus is so designed as to allow ample margin for excess power if it should be needed. The efficiency of the apparatus will not be appreciably affected by a considerable variation in the maximum power required. The figures here given apply to a vacuum of only 28 inches, but experience with condensing apparatus of such size as that proposed by the Department for this vessel would indicate that a much better vacuum should almost always be carried. Improvement of vacuum will effect a diminution of steam consumption at all speeds at the rate of about $\frac{3}{4}$ pound per

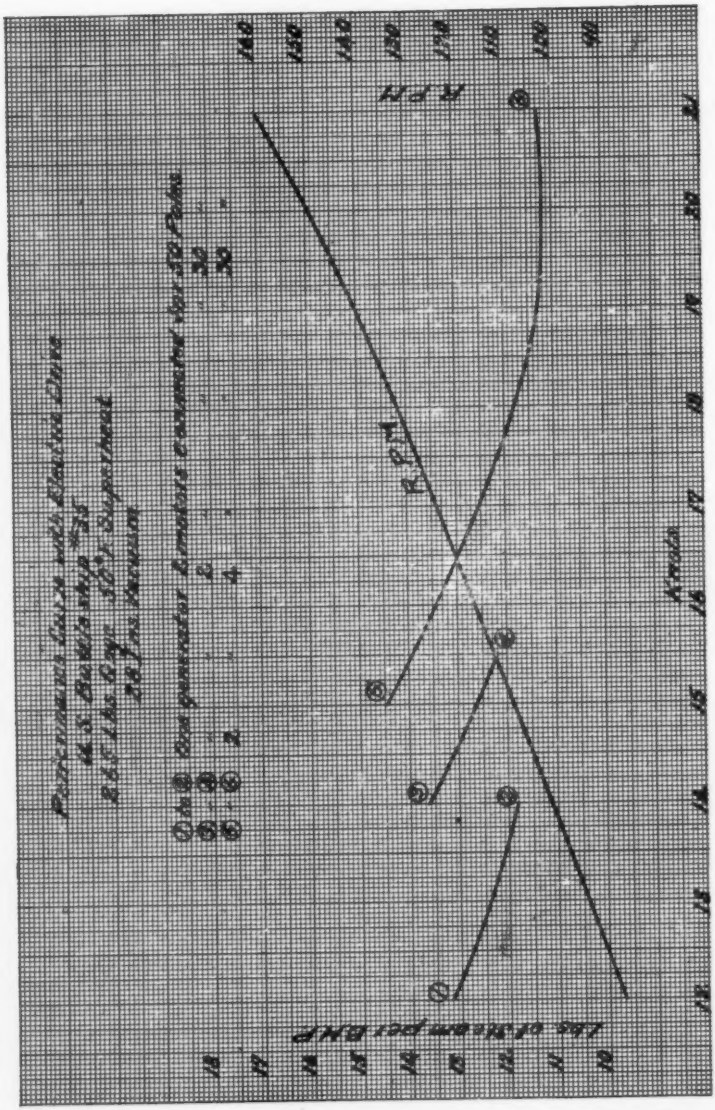


Plate III.

inch. In many plants on shore, using similar turbine apparatus and no greater condensing surface, the vacuum averages over 29 inches throughout the year, and in such a vessel it is believed that similar results are attainable.

STEAM PRESSURE, 265 POUNDS GAGE; VACUUM, 28 INCHES; SUPERHEAT, 50 DEGREES.

Knots.....	12	16	18	19	20	21
Motor speed.....	87	117	132	140	149	160
Generator speed...	1,110	895	1,010	1,072	1,140	1,220
Shaft brake H.P..	4,630	11,100	16,400	19,400	23,100	29,000
Pounds steam per shaft H.P.....	13.2	13.4	11.8	11.3	11.25	11.3

REPAIRS.

The generator and turbines can be uncoupled from each other and lifted separately and the pairs of motors can similarly be uncoupled. It would thus be relatively easy to move any part which might require repair. Spare windings and spare turbine buckets can easily be carried and the operation of introducing them would not be very difficult. All parts of the turbine, except the lower stationary blades, can be got at by simply lifting off the top, and there is no damage to these stationary blades which could do more than impair the economy of the turbine which will always be in condition to give service when the rotating element is intact. In this connection it should further be remembered that either one of the four motors can be used to propel the ship with either one of the two generating units, so that the possibility of complete disablement is extremely remote. All the valve mechanism, bearings, switching apparatus and instruments are readily accessible for repair or replacement, and the conditions of operation are such that the ship could easily be run with temporary connections and hand-steam admission, even if all the governing apparatus and switching mechanism were out of order. Such adaptability constitutes one of the important advantages of all electric transmission and should

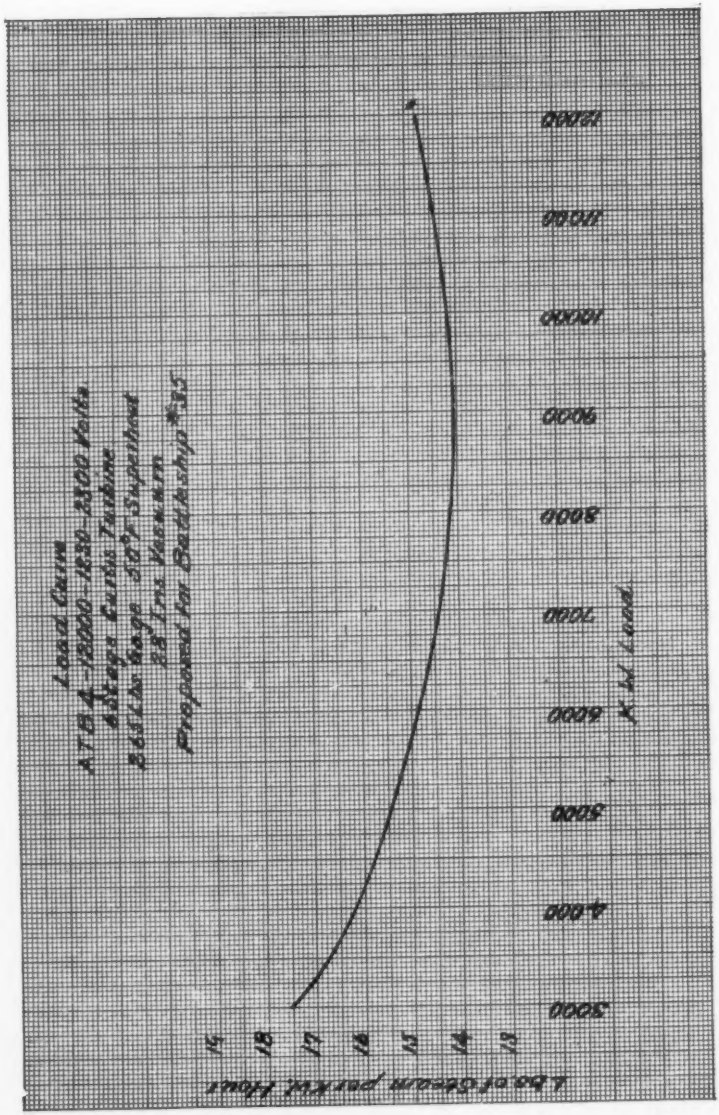

Load Curve
AT 68 A.-12000-1250-2300 Volts
4 Kings Curtis Turbine
265 Lbs. Gage 30° Superheat
28" Ins. Vacuum
Proposed for Battleship #35
Lbs. of Oil per K.W. Hour
15
16
17
18
19
K.W. Load
5000
4000
6000
7000
8000
9000
10000
11000
12000

Plate IV.

never be lost sight of in such an engineering enterprize. The electrical circuit can always be quickly established, whereas a mechanical connection, if wrecked, generally necessitates reconstruction.

ARRANGEMENT IN SHIP.

The accompanying drawing (Plate I) shows the apparatus here proposed installed in the engine room of vessel as designed for the installation of Curtis turbines, with all the auxiliary devices proposed and shown in the Department's designs for such a Curtis installation. If desirable, the arrangement shown could be changed in many ways, and it is quite probable that a further study of all needs might lead to some unimportant alterations. This drawing, however, serves to illustrate the practicability of the installation proposed. The drawing does not show all the piping, but the arrangement seems to afford a considerable saving in weight and complication of piping and to give a less crowded condition. This installation requires no appreciable change in the design of engine room adapted for the Curtis turbine and, except for the matter of propeller speed and consequent shaft diameter, the two installations might be made interchangeable without appreciable change of the auxiliary apparatus.

Excitation.—The plan proposed contemplates excitation of the generators from a separate fixed potential source, the exciting current so delivered being controlled by rheostats. When both generators are in use a maximum exciting power of about 140 kw. will be required, and when one generator is in use the maximum will amount to about 70 kw. This power can most advantageously be derived from the generating plant of the ship, but might, if for any reason desirable, be taken from independent steam-driven units. Since connection to the ship's circuits, will be the best method, no exciting apparatus has been included in this proposition.

WEIGHTS.

	Pounds.
Two generating units	674,000
Accessories	2,600
Four motors	408,000
Switchboard and switches	5,000
Field rheostats	1,600
Water-cooled rheostats	6,000
Cables	17,800
Total weight	1,115,000

ADVANTAGES.

The advantages of the installation here described may be summarized as follows:

1st. The full-speed economy would be superior to that of any turbine-driven ship because the high-speed turbine is much more efficient and because the propellers operate at speeds which will give much better propulsive efficiency. The difference of these propulsive coefficients, as indicated by the data in possession of the Navy Department, would just about counterbalance all the electrical losses.

2d. The economy at cruising speeds will be very far superior to any method of propulsion applied to battleships, being, as shown by the accompanying table and curve, almost equal to the economy at full speed. This is accomplished through arrangements by which only part of the apparatus is used at low speed, and by which the turbine at such times can be worked at an economical rate of revolution.

3d. The repairs of this installation could be much more easily made than with direct turbine drive, all the parts being lighter and more accessible.

4th. Reversing is accomplished without any special reversing wheels, reversing power can be delivered very quickly, and the torque available will be greater with the same steam consumption than that which is generally available in turbine-driven ships.

5th. The turbines used are far simpler than those applied to direct propulsion, and the electrical apparatus is simpler, more dependable and more easily repaired than any turbine or possible form of steam prime mover.

APPARATUS PROPOSED FOR NAVAL COLLIER.

The equipment proposed for a collier is similar in principle of action to that above described, but, owing to the different character of service, can be made simpler. In this case one generating unit and two motors would be installed. The motors would have no arrangement for pole-changing, but would be equipped with resistance attachments in a manner similar to that above described. In the case of the collier installation it is proposed to install oil switches to connect the different parts of the apparatus, which switches are so designed that they can be opened or closed under any conditions of load. This would constitute a simplification which can be conveniently applied in this case. In the battleship such oil switches would take up a good deal of room, and, since they there afford no important advantage, it has been thought better to save space and operate in the manner described, by interrupting the circuit when the connections are changed.

In the case of the collier installation the ventilation of the apparatus could be taken care of in the same manner described for the battleship. It was, however, proposed that in this case a slightly different arrangement be adopted. The motors would be installed in such a manner that their ventilating air would be drawn from the engine room and that the air discharge from the motors would be delivered to the suction of blowers used in connection with the Howden-draft system for the after boiler room. Similar delivery of heated air to the ship's furnaces could be provided for in any other electrically-propelled ship, and would effect appreciable fuel saving. It is not, however, in any way necessary to the success of the installation. Separate ducts being provided to deliver cool air to the electrical apparatus and to convey the heated

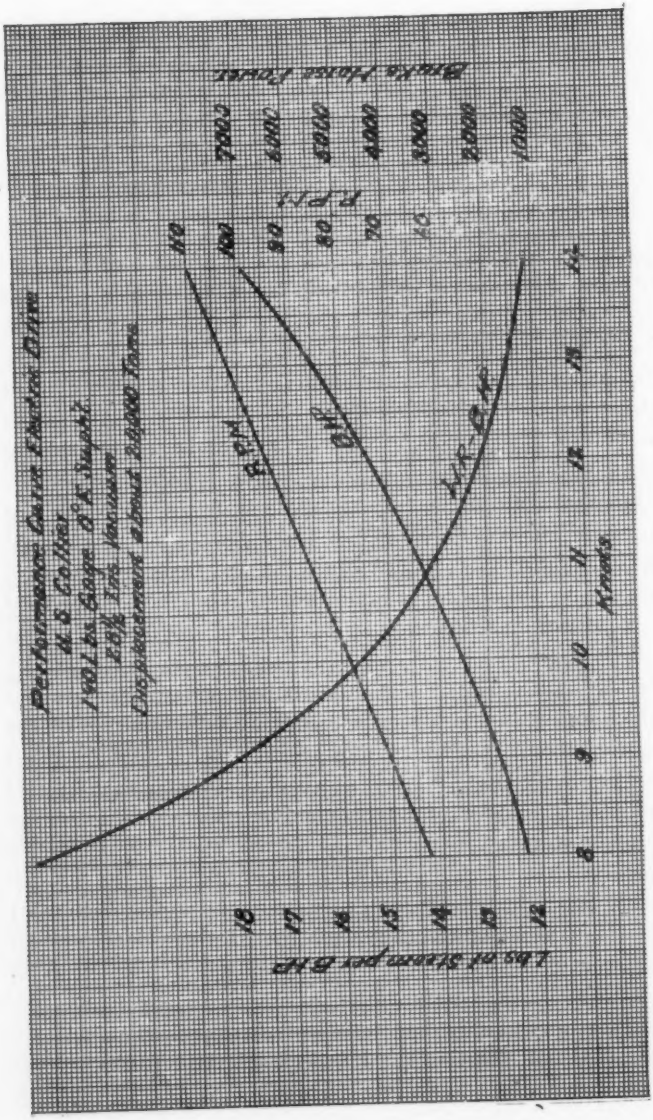

Performance Curve Electric Drive
U.S. Collier
160 ft. B.P. Length
Displacement about 16000 Tons.
Brake Horse Power
7000
6000
5000
4000
3000
2000
1000
R.P.M.
Lbs. of Steam per B.H.P.
Knots

Plate V.

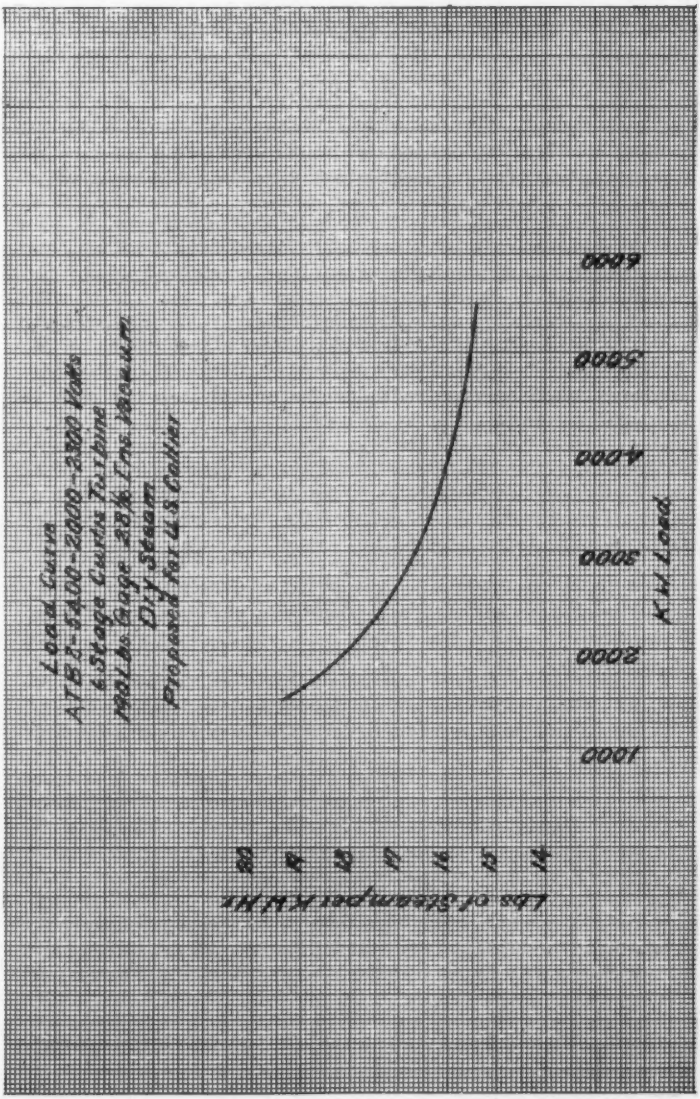

Load Curve
AT&T-5400-2000-2300 Volts
6 Stage Curtis Turbine
190 Lbs. Gage 28½ Ins. Vacuum
D.V. Steam
Proposed For U.S. Collier
K.W. Load
6000
5000
4000
3000
2000
1000
Lbs. of Steam per K.W.Hr.
8
7
6
5
4

Plate VI.

air away, there is no escape of such heated air into the engine room, and the temperature there will be no higher than with other methods of propulsion.

When resistances are connected the reversal of the vessel would be accomplished by simply throwing the switch levers in either direction as desired, without interrupting the circuit or touching the steam connections. Such convenient reversal of either propeller would be an advantage in such a vessel since large freighters do not steer very well at low speed, so that instantaneous manipulation of the propellers would be very convenient in anchoring, docking, etc.

Economy.—The following table and the accompanying curves show the results which can be accomplished with this apparatus when operated with dry saturated steam having a pressure of 190 pounds gage and with a vacuum of 28.5 inches:

Knots	10	12	14
Motor speed	78.5	94.5	110
Generator speed	1,430	1,725	2,000
Estimated shaft H. P.	2,550	4,350	6,900
Water rate in pounds per brake H.P.	15.55	13.15	12.15

These calculations are based upon the assumption that the power required will be in accordance with the figures given in this table. Provision is made, however, for a considerable larger capacity if it should be required, and the efficiency of the processes would not be materially affected if the demand for power were considerably larger or smaller.

The writer has no means of justly comparing the economy of the performances here shown with those of the best existing apparatus on similar vessels. Available figures, however, indicate some saving in fuel consumption, and there is a saving in weight equal to about 1½ per cent. of the carrying capacity of the vessel.

Such a vessel as this collier does not afford a very favorable

case for electric propulsion, the purpose of the proposed installation being largely to demonstrate to the Navy Department the practicability of the methods used. The uses to which these methods are best adapted are the propulsion of large war vessels and large moderate-speed liners.

WEIGHTS.	Pounds.
Turbo-generating unit	168,000
Two induction motors	135,600
Switchboard and motor panels	4,100
Two water-cooled resistances	3,000
Cables	3,500
Turbo-driven exciter	3,800
Total weight	318,000

Arrangement in Ship.—The accompanying drawing (Plate II) shows the proposed arrangement of this apparatus in the ship. The positions of the propeller shafts adopted are the same as those provided for engine drive. The only change necessary is to provide pockets in one or two of the floor plates to make room for the lower part of the motors.

Advantages.—The advantages of this collier installation as compared with the reciprocating engine designed for this ship are:

1st. The weight is greatly reduced.

2d. The economy is better.

3d. No oil is lost and none gets into the steam.

4th. The labor and attendance would be greatly reduced.

5th. The apparatus is simpler and more easily maintained.

Assuming 14 pounds per brake H.P. as water rate with engine drive, the improved economy combined with the reduced weight would enable the ship to carry 9 per cent. more coal if she had to deliver her cargo at the end of a thirty-day voyage and return in ballast.

In a lecture delivered by the Engineer-in-Chief of the Navy before the Naval War College on August 9th, 1910 (J. A. S.

N. E., vol. xxii, No. 4), he suggests the following points as probable subjects for special investigation in connection with such an equipment.

(a) Weight, as compared with all-turbine drive.

(b) Ventilation of the generators and motors, there being considerable heat generated in the armatures.

(c) Cooling of resistances. Under certain conditions there are large quantities of heat to be suddenly dissipated from the resistances.

(d) The possible failure of the machines or parts of the machines due to access of water.

(e) Complexity of accessories—switches, pole-changing devices, resistances, etc.

(f) The degree of delicacy of speed control.

(g) The effect on chronometers, compasses and personnel of grounds, the tension being in excess of 2,000 volts.

All such matters should be thoroughly investigated in connection with the designing of every important engineering undertaking, and most of the matters here mentioned can be thoroughly investigated in connection with existing installations of similar kinds of apparatus on shore.

The resistances mentioned under (c) cannot be so investigated, but extensive experimenting has been done for the special purpose of establishing the limits of safe application in such designs, and a study of these experiments will afford conclusive proof of practicability.

Item (f), the degree of delicacy of speed control, will certainly not need special investigation, since the speed control in this case is by simple variation of the steam admission and is therefore similar to that of every other turbine-driven vessel.

Under (g) two points are raised, one being the matter of safety to life with 2,000-volt windings, and the other being possible effect upon magnetic conditions in the ship. With this installation the windings which transmit the power are both attached to stationary elements, the armature of the generator as well as that of the motor being stationary. These

windings are entirely enclosed and protected so that they could not possibly be got at while the apparatus was in operation. The leads from the generators and the switches will be heavily insulated and entirely out of reach of any contact with a man's body. The switches are enclosed in such a manner that none of the current-carrying parts can be touched unless the switch is taken apart. To get a shock, therefore, with such an equipment it would be necessary for a man to deliberately take the apparatus apart while it was running or else cut through the insulation of some conductor, which is installed out of his reach or which can, if desired, be put behind a protecting barrier or in a metal duct.

With such apparatus there could be no effect upon magnetic conditions in the ship since the magnetic circuits of such apparatus are entirely interior, no magnetic field whatever existing outside of the apparatus itself. In this respect such apparatus would be much less active than the direct-current apparatus now used on war vessels, and the circuits from it carrying multi-phase alternating currents would be completely neutral.

THE ECONOMIC ADVANTAGES OF THE REDUCTION GEAR AS APPLIED IN THE PROPULSION OF NAVAL VESSELS.

By Lieut. W. W. Smith, Member.

In naval vessels, economy in fuel has always been important, but it seems that its extreme importance has not been fully realized until lately; for it has only been recently that such strenuous efforts, induced by keen competition, have been made to realize the highest attainable economy in regular service through efficient operation. As to the correctness of this policy, there is no question but, without detracting from the importance of operation, too much should not be expected from it; for it should be remembered that the design sets a limit beyond which no further economic gain is possible, no matter how carefully and intelligently the machinery may be operated. Therefore, in the endeavor to obtain high fuel economy in service, the importance of the inherent economy of the machinery is apparent.

As the steaming fuel economy depends principally on the economy of the main engines, which at present is not as high as desirable, a material increase in engine economy would result in a substantial increase in fuel economy. The purpose of this paper is to show that present fuel economies can be increased materially by the use of the high-speed turbine and speed-reduction gear, and, to point out the effect of this increase in economy on the naval qualities of the vessel.

The fuel consumption for the propulsion of a vessel depends on the efficiency of generating steam and the quantity of steam used by the propelling machinery—both main and auxiliary. It is largely determined by the steam consumed by the main engines, and, as the scope of this paper does not include the boilers and auxiliary machinery, they are elimi-

nated by assuming a standard performance. With this assumption, the fuel consumption is determined by the steam consumed by the main engines which depends on the efficiencies of the engine and of the propeller, or on the over-all efficiency of the propelling machinery.

If all propellers were of the same efficiency, or, even nearly so, comparative fuel consumptions could be represented by the water rates per S.H.P. of the engines, as is frequently done. As, however, propellers have a wide variation of efficiency, depending principally on the type of propelling machinery, it is evident that such a comparison would not be correct, except where the efficiencies are the same, and, in comparing different types of machinery, the error would frequently be as much as 15 per cent.

As the fuel consumption depends on the over-all economy, the correct and logical method is to represent it by the over-all economy—the water rate per E.H.P. : a method of general application by which all types, even though widely different, can be compared. This method of expressing the over-all economy of the propelling machinery in terms of water rate per E.H.P. is similar to the familiar way of expressing the over-all economy of an electric-generator unit in terms of water rate per kilowatt.

Frequently, there is doubt as to the accuracy of the S.H.P. and, consequently, the propulsive efficiency. However, errors in the S.H.P. and propulsive efficiency do not affect the correctness of the water rate per E.H.P., provided, the total water used by the turbines and the E.H.P. are correct.

For the curves given herein, as noted below, the water rate per E.H.P., W_e, was obtained from the water rate per S.H.P., W_s, and the propulsive efficiency, P, as follows: $W_e = \frac{W_s}{P}$. With the above assumption, it is obvious that the errors in W_s and in P balance each other and that both may thus be incorrect without affecting the correctness of W_e; for $W_e = \frac{W_s}{P} = \frac{W}{S} \times \frac{1}{P} = \frac{W}{S} \times \frac{S}{E} = \frac{W}{E}$, where $S =$ the

shaft horsepower of the propeller shaft, E = the effective horsepower delivered by the propeller and W = the total steam per hour used by the turbine.

There may be some doubt as to the accuracy of the E.H.P. determined by the present method of towing a model of the vessel and calculating the E.H.P. by the law of comparison; but, as it is the only method employed, it is necessary to assume its correctness for the data given.

Because of the importance of determining accurately the over-all economy of the propelling machinery, the actual E.H.P. developed should be measured. It would not be exceedingly difficult to measure the thrust of the propeller; from which, the speed being known, the E.H.P. could be readily calculated. The present method of standardizing is well suited for determining the curve of E.H.P.—speed, which is of sufficient importance to warrant the slight expense necessary.

The uses of the E.H.P.'s determined on trial and in the experimental model basin should not be confused, as it is not intended to detract from the importance of the latter. The experimentally determined E.H.P. would be used as a preliminary determination for obtaining the power required. The measured E.H.P. would be used for the accurate determination of the performance of the propelling machinery: information which would be of great value to the designer in his efforts to improve economy, and, of still further value in the experimental determination of the E.H.P.

It is assumed that all vessels cruise with the fleet and that the ordinary cruising speeds for all vessels will be between 10 and 15 knots. As indicated in the curves, the cruising speeds are taken as follows: low cruising speed, 10 knots; medium cruising speed, 12½ knots; fast cruising speed, 15 knots.

The range of speed, as indicated, extends from the low cruising speed of 10 knots to the full speed, and, to make clear its effect on economy, the curves were laid out on per cent. of full speed. As the cruising speeds are the same for

all vessels, it follows that, the greater the speed, the greater is the range of speed.

The horizontal line, EE, in Fig. 1, corresponding to 25 pounds water rate per E.H.P., was arbitrarily taken to represent the limit of good economy and, therefore, the range of good economy is the range of speed corresponding to the portion of the curve that lies below this line.

The performances of several types of propelling machinery are given in the curves and tables. The cruising speed, range of speed and limit of good economy are taken as explained above. The water rate per E.H.P. was obtained from the water rate per S.H.P. and propulsive efficiency as follows: $W_e = \frac{W_s}{P}$.

The comparisons are based on the over-all economy, which is expressed by the water rate per E.H.P.

As the main propelling machinery (main turbines and propellers) of the destroyers is the same for both the direct-connected and reduction-gear cruising turbines, the propulsive efficiencies are the same; and, as the economies of the turbines are proportional to the over-all economies, the comparative results being the same, the former was taken to represent the latter.

The S.H.P. delivered to the propeller shaft was used in all cases to obtain the water rate per S.H.P. In the geared turbine, therefore, the efficiency of the gear is included in its economy.

The use of cruising turbines in connection with the main reduction-gear turbines is apparent from the curves. Although the curves do not make it plain, the performances of the direct-connected reaction-type turbines include the cruising-turbine combinations for medium and low speeds. The impulse turbines given are not fitted with cruising turbines. The curves, therefore, represent the best performances at all speeds.

(A) The curves, Fig. 1, for the 25-knot battleship represent good performances of the direct-connected and reduction-gear turbine machinery, and are characteristic of these types.

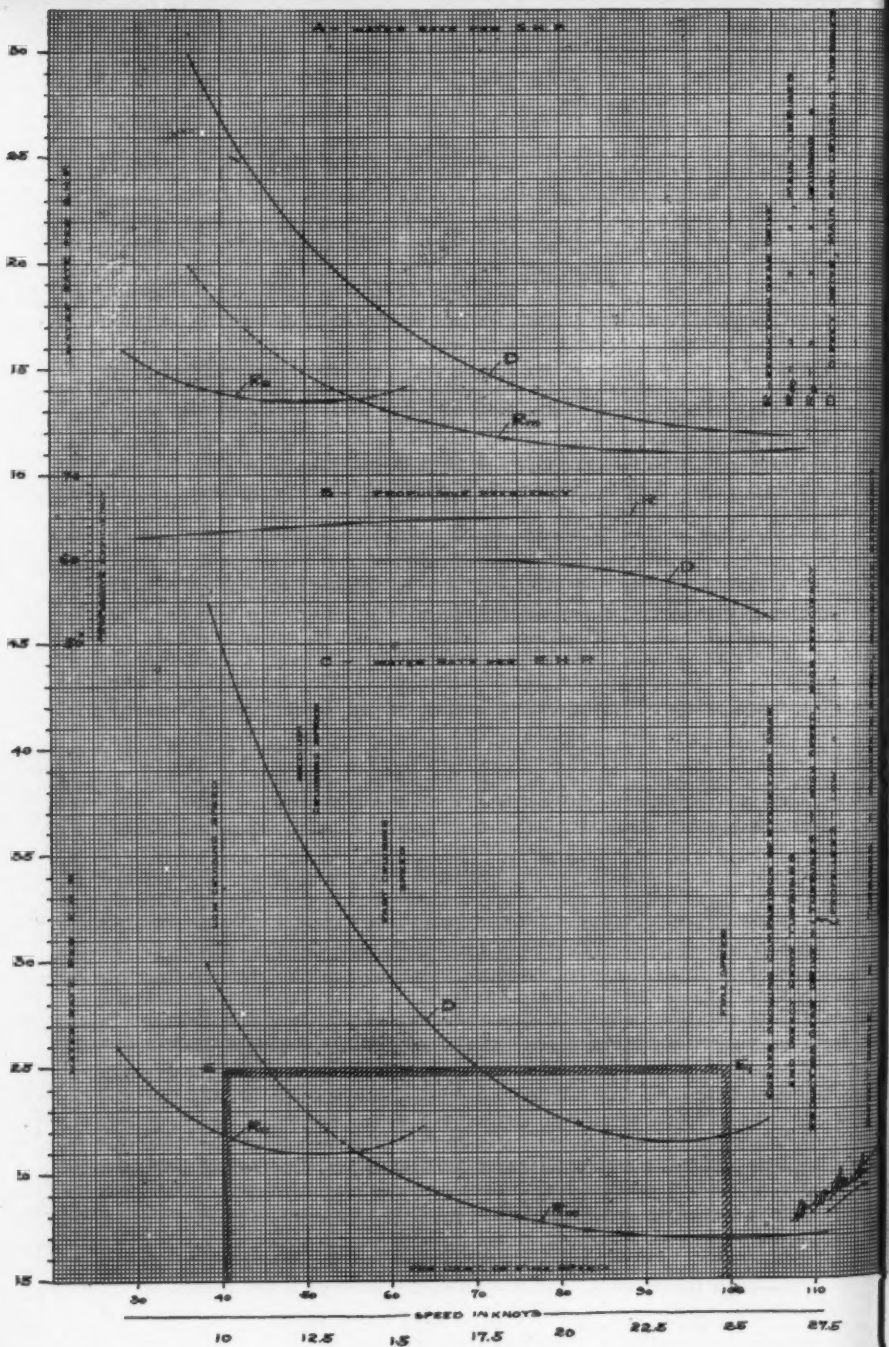


Fig. 1.—A.

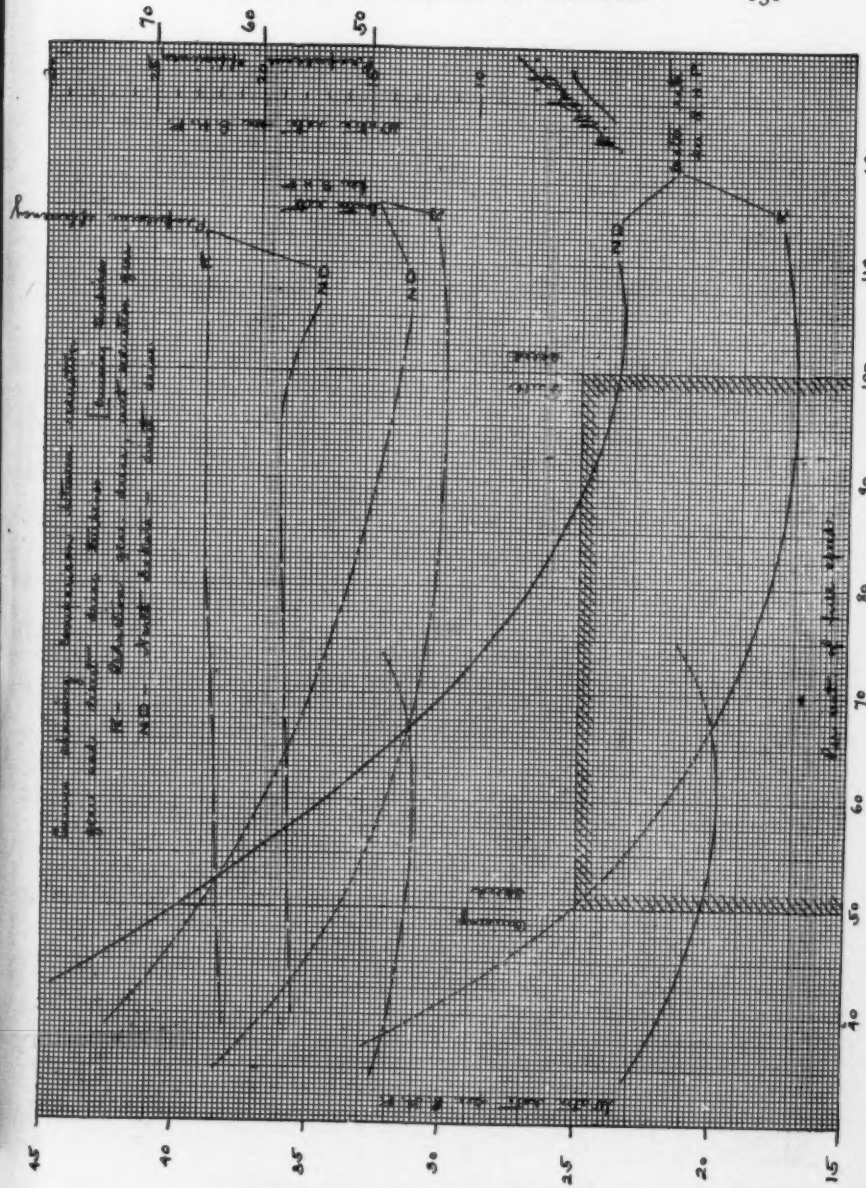


Fig. 2.—B. Full Speed, 21 Knots.

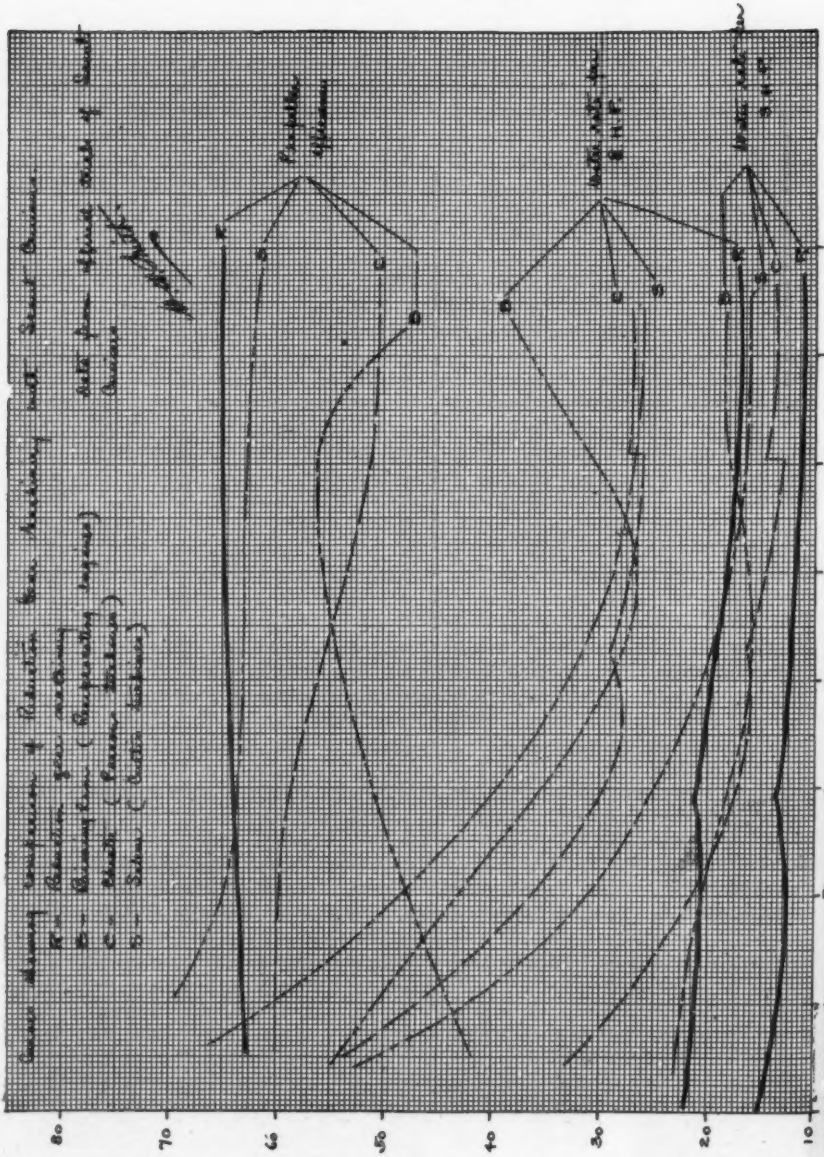


Fig. 3.—C. Water Rate per S.H.P. and E.H.P. and Propeller Efficiency.

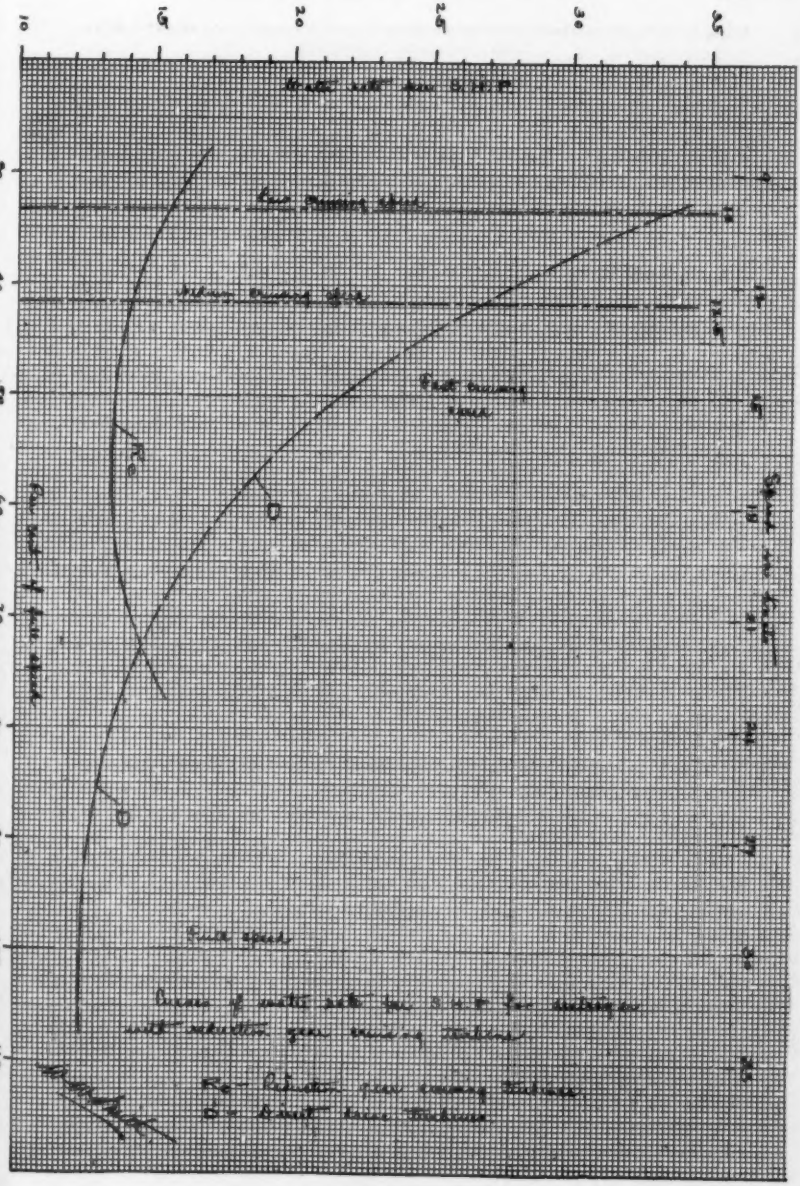


Fig. 4.—D.

TABLE I.—COMPARISON OF REDUCTION-GEAR AND DIRECT-DRIVE MAIN AND CRUISING TURBINES.

(Water rates per E.H.P.)

	Ship.	Speed.	Per cent of full speed.	Water rate per E.H.P.		$\frac{D}{R}$
				Direct-drive D.	Reduct'n gear drive R.	
A	25-knot battleship.....	25	100	21.8	16.9	1.29
		15	60	29.2	20.1	1.45
		12.5	50	35.2	21.0	1.68
		10	40	45.0	21.9	2.05
B	U. S. S. *North Dakota*..... ...	21	100	23.6	17.0	1.39
		15	72	29.4	19.2	1.53
		12.5	60	34.6	19.8	1.75
		10	48	41.4	20.7	2.00
C	U. S. S. *Salem*.....	25	100	26.0	17.0	1.53
		15	60	39.4	20.8	1.89
		12.5	50	48.5	20.5	2.36
		10	40	60.0	21.1	2.84
C	U. S. S. *Chester*.....	25	100	26.9	17.0	1.58
		15	60	29.6	20.8	1.42
		12.5	50	36.3	20.5	1.77
		10	40	46.8	21.1	2.21
C	U. S. S. *Birmingham*.....	25	100	36.2	17.0	2.13
		15	60	35.3	20.8	1.70
		12.5	50	43.0	20.5	2.10
		10	40	50.6	21.1	2.39

TABLE II.—COMPARISON OF REDUCTION-GEAR AND DIRECT-DRIVE CRUISING TURBINES.

(Water rates per S.H.P.)

	Ship.	Speed.	Per cent. of full speed.	Water rate per S.H.P.		$\frac{D}{R}$
				Direct-drive D.	Reduct'n-gear drive R.	
D	30-knot destroyer.....	15	50	21.7	13.6	1.60
		12.5	42	26.8	14.2	1.88
		10	33	34.5	15.5	2.22
E	U. S. S. *Flusser*.....	15	50	23.8	13.6	1.76
		12.5	42	28.5	14.2	2.00
		10	33	35.6	15.5	2.30

The performance of the direct-connected turbine is slightly better than the best performance of the reaction type available. A comparison of the latter with the best results of the impulse type shows: For the impulse type, the water rates per S.H.P. are greater, the propulsive efficiencies are greater; and for both types, the over-all economies are about the same. Although there may be a slight difference between the two types, it will be sufficiently accurate to assume that the curve of water rate per E.H.P. represents a good performance for the direct-connected turbine.

From the test results of the reduction gear the efficiencies at high and low speed were taken as 96 and 98 per cent., respectively; somewhat lower than the test values. From the test results of numerous high-speed turbines, probable good performances were assumed for obtaining the curve of water rate per S.H.P. The propulsive efficiency was taken as 65 per cent. at full speed, which should easily be obtained, and slightly less at the lower speeds. It is believed that the over-all economy, obtained from these two assumed curves, is a fairly close approximation to the results that could be obtained with this type of machinery.

The characteristics of these curves, which are referred to later, should be noted:

(B) The performance data of the *North Dakota* were obtained from the Journal of the Society.

(C) The data for the scout cruisers were obtained from the report on the comparative trials published by the Navy Department, December 22, 1909.

(D) The curves, Fig. 4, for the 30-knot destroyer represent good performances of direct-connected turbines in connection with direct-connected cruising and reduction-gear cruising turbines, and are characteristic of these types, the data for these types of machinery being assumed in the same manner as described for "A."

(E) The data for the *Flusser* were obtained from the Journal of the Society.

Although it is probable that the reciprocating engine will

not be used in future large high-speed vessels, it is of interest to note the best results that have been obtained. On the *Delaware*, the water rates per I.H.P. were: At full speed, 13.4 pounds; at 89 per cent. of full speed, 13.0, and at 57 per cent., 15.32. The probable values per S.H.P. and per E.H.P. are: 14.2, 13.8, 16.2 and 22, 21, 24.

The engine economy of the *Delaware* is better at all speeds than many turbine vessels. At full speed the best turbine results are not much better, and at low speeds they are considerably worse. As far as the economy alone is concerned, the reciprocating engine is far from being a "dead letter."

There are other reasons, however, which make the turbine desirable for naval vessels; some of the reasons are: Ability to maintain the full speed for long periods; readiness for service after long periods of steaming; and, lack of vibration, giving a more steady gun platform.

The economy of the turbine at full speed is good, we may say satisfactory, but at low speeds it is not. The reasons for this are:

The turbine is inherently a high-speed machine, in which the range of good economy is limited. On the other hand, a comparatively low speed is essential for the propeller, which has a comparatively large range of good economy. To reconcile the high speed of the turbine with the low speed of the propeller, it is necessary to compromise both; and, as neither runs at the speed for which it is suited, the efficiency of each is less than it would be at what may be called "its normal speed." The extent of the compromise depends principally on the speed; the lower the speed, the greater is the compromise.

As the turbine is under-speeded, the range of good economy is small as compared with the normal range of a high-speed turbine. To improve this quality, cruising turbines are used in some cases; but, even with cruising turbines, the low speed is so adverse that the range of good economy is not increased to the extent desirable.

As the propeller is over-speeded at full speed, its efficiency is decreased. At lower speeds, however, the efficiency improves.

If we seek the ideal conditions for the turbine and propeller individually, we will find them to be:

The turbine would run at the high speed required, with most of its range of good economy below the full speed. The required range of speed may exceed the range of good economy, as in high-speed vessels, in which case, a second turbine, also running at high speed, would be used, its range of good economy extending from the lowest speed of the former to the lower limit of speed, so that the combined range of good economy would cover the entire range of speed.

The propeller would run at the low speed required and its range of good efficiency would cover the range of speed.

To attain these ideal conditions, the desirability of a speed-reduction gear is obvious. It is clear that the efficiency of transmission should be high, and that the most efficient, if otherwise satisfactory, will be the best. If we assume that the spur-wheel reduction gear is otherwise satisfactory, as has been fairly well demonstrated, it is evident that its superior efficiency precludes other known systems.

As the speed of the vessel increases, the normal speeds of the turbine and propeller approach each other and, at some high speed, they become the same. At this speed, the advantages of the reduction-gear disappear and, above this limit, the turbines should be direct connected. As the large range of speed would exceed the range of good economy, a high-speed geared turbine would be used to extend the range of good economy to the lower limit of speed.

The speed at which the advantages of the reduction gear disappear is not known—it will probably vary somewhat with the type of vessel—but it appears to be well above 30 knots. It is interesting to note that, for a 33-knot destroyer (H. M. S. *Mohawk*), it was found that the advantage was still in favor of the reduction-gear machinery.

The over-all economy of direct-connected turbine machinery will undoubtedly be improved, but it is doubtful if it will ever equal that of the high-speed turbine machinery, except

at high speeds. It is safe to say that the high-speed turbine machinery will always give the highest over-all economy at medium and low speeds, both for main and cruising turbines.

The use of cruising turbines depends entirely on the range of speed and the range of good economy. If the latter is less than the former, cruising turbines should be used. It is an advantage to avoid the complication of cruising turbines and, as it is probable that the range of good economy will be increased in future turbines, they may not be necessary in any but high-speed vessels where the range of speed is great.

For naval vessels in general, the speeds may be classified as follows: high speed, above 25 knots; medium, from 15 to 25; and low, below 15. The first is represented by the destroyer; the second by the cruiser and battleship; and the third by the fleet auxiliary. This general classification referred to above should not be confused with that for a particular vessel or type.

The naval requirements for the propelling machinery are that it will give the greatest speed and steaming radii. As the economy, particularly of turbine machinery, decreases at low speeds, these requirements become speed and steaming radius at cruising speed.

With a given design of machinery, the speed is limited by the weight of the machinery and the cruising radius by the weight of coal. In this case, therefore, the naval qualities are determined by the weight of machinery and coal.

With a given weight, the speed is determined by the effective power that can be developed with the given weight of machinery and the steaming radius by the effective power that can be developed with a given weight of coal.

With a given design of boilers and auxiliary machinery necessary for propulsion, which, for convenience, will be called the "other machinery," their weight will be determined by the quantity of steam required for the main propelling engines at full speed; and, therefore, with a given vessel, the weight of the other machinery is determined by the over-all economy of the engines and propellers, which will be called the "propelling machinery." With a given weight of coal,

it is obvious that the cruising radius depends on the economy of the propelling machinery. Therefore, if the weight of the propelling machinery is neglected, the naval qualities depend entirely on its economy.

The following example illustrates the effect of the economy of the propelling machinery on the naval qualities of a vessel:

Take the 25-knot battleship "A," the curves for which are given in Fig. 1.

Given the full speed and cruising radius; to determine the naval qualities of the D (direct-drive) and the R (reduction-gear drive) types of machinery.

The weight and performance of the other machinery for a given quantity of steam supplied to the propelling machinery is the same for both.

To show that the weight of the propelling machinery has little effect on the naval qualities as compared with its economy, the weight of the R type is taken 20 per cent. greater than D.

S = Full speed in knots = 25.

S_c = Cruising speed in knots = 10.

E_f = E.H.P. at full speed = 20,000.

E_c = E.H.P. at cruising speed = 1,500.

W_f = Water rate per E.H.P. at full speed in pounds per hour. For D, $W_f = 21.8$. For R, $W_f = 16.9$.

W_c = Water rate per E.H.P. at cruising speed in pounds per hour. For D, $W_c = 45.0$. For R, $W_c = 21.9$.

O = Weight of other machinery in pounds per pound of steam supplied to the propelling machinery at full speed = 7.0.

C = Weight of coal in pounds required to generate one pound of steam for the propelling machinery at cruis-speed = .14

M = Steaming radius at cruising speed in nautical miles = 10,000.

P = Weight of propelling machinery in pounds per effective horsepower at full speed. For D, $P = 50$. For R, $P = 60$.

Weight in tons of other machinery required $= \frac{O \times W_f \times E_f}{2{,}240}$.

For D, = 1,360. For R, = 1,055.

Weight in tons of coal required $= \frac{C \times W_c \times E_c \times M}{S_c \times 2{,}240}$.

For D, = 4,220. For R, = 2,050.

Weight in tons of propelling machinery $= \frac{P \times E_f}{2{,}240}$.

For D, = 447. For R, = 535.

Weights in tons for—	D and R	
Propelling machinery,	447	535
Other machinery,	1,360	1,055
Coal,	4,220	2,050
Total,	6,027	3,640

Or, taking one-half of the weight of the coal, the totals are 3,917 for D and 2,615 for R.

If the example had been: given the weight, to determine the speed and cruising radius; it is evident that the R type would excel in both.

In the example, it is clear that the weight of the propelling machinery has little effect on naval qualities as compared with its economy. Although the weight taken for the R type is considerably greater than its probable value, it is evident that it would still retain its advantage, even if a very much greater value were taken.

It is obvious that the naval qualities depend almost entirely on the economy, and, therefore, in designing or selecting the propelling machinery, it should be the primary consideration.

Referring to the characteristic curves A, Fig. 1, for the battleship, it will be seen that the low-speed turbine economy decreases rapidly at low speeds and that the high-speed propeller efficiency decreases near the full speed. The results of the compromise of speeds and the inimical effect on the overall economy caused by the best performance of one being counteracted by the worst of the other is apparent.

As compared with the above, the high-speed turbine econ-

omy is better, particularly at low speeds; the low-speed propeller efficiency is better, particularly near the full speed; and, as the performances of both are better, the combined performances or the over-all economy is much better, particularly at low speeds.

The range of good economy of the latter covers the entire range of speed, while that of the former covers but half of it.

The over-all steam consumptions of the direct-drive machinery at full and low speed are, respectively, 1.29 and 2.05 times those of the reduction-gear machinery; and these are also the expressions for the relative fuel consumptions.

It is clear that the superior naval qualities, resulting from the better economy of the reduction-gear machinery, makes it the more desirable.

Referring to the characteristic curves D, Fig. 4, for the destroyer, the superiority of the reduction-gear cruising machinery is apparent.

As the foregoing remarks also apply to the other curves, the comparisons should be quite clear.

As they have been considerably improved in later vessels, the performances of the scout cruisers should not be regarded as representative; they are given as a matter of interest only.

The comparative advantages of the reduction-gear machinery, as clearly expressed by Mr. George Westinghouse, are of interest:

"However much the new system promises for express steamers in the mercantile marine, it has vastly more advantages as applied to naval vessels. The express steamer normally runs at its highest speed, and this is the condition for maximum turbine efficiency. This is especially true in the case of turbines connected directly to the propeller shaft, for the reason that, as outlined before, the peripheral speeds and number of rows of blades are at best below the requirements of efficient design, and any dropping below the maximum working speed accentuates the bad effect of the deficiency.

"On the other hand, in the case of a battleship or a cruiser, maximum speed is only an emergency condition. The nor-

mal cruising speed is only about 60 per cent. of the maximum speed, and requires, perhaps, less than 25 per cent. of the maximum power. It is at cruising speed that turbine-propelled naval vessels have shown to disadvantage, as compared with vessels propelled by the best types of reciprocating engines. By reason of the more liberal blading that is possible in a high-speed turbine, its economic performance is less sensitive to departures from the maximum rotative speed, than is that of the low-speed turbine.

"The United States Government has lately awarded contracts for two new battleships, to be equipped with steam turbines. These battleships are to have a speed of 20½ knots, which will require in round numbers 28,000 shaft horsepower. With 55 per cent. propeller efficiency, the effective propelling power will be about 15,400 horsepower. With 65 per cent. propeller efficiency that is easily possible with the reduction gear and a propeller at a lower speed of revolution, this same propelling power would require less than 24,000 horsepower on the shaft. The average steam consumption guaranteed at full power is about 14½ pounds per shaft horsepower. With the better steam economy of the high-speed turbine, the boiler capacity required would be reduced fully one-third. With the same bunker capacity, the radius of action would be enormously increased, which is an advantage of incalculable value.

"If the same boiler equipment as is now proposed were maintained, there would still be a saving in weight of over 250 tons, or approximately one-eighth of the total penalty weight of the machinery in each ship, resulting solely from the substitution of the high-speed turbine and reduction gear for the more cumbersome slow-speed direct-connected machine. At the same time, by reason of the well-known overload capacity of a liberally-proportioned turbine, there would be available a surplus power of about 50 per cent., which should make possible an emergency speed of nearly 3 knots in excess of that called for in the specifications."

As nearly all steaming is done at low speeds, it is safe to say that, taken over a considerable period—a year, for example—

the saving in steaming fuel effected by the reduction-gear machinery would be from 30 to 40 per cent. When the large amount of money expended for the steaming coal of the fleet during a year is considered, the future possibilities of the machinery that will make such a large saving possible will be more fully appreciated. Financial saving in the maintenance of a large fleet is of great importance, both in times of peace and war; but, when opposed to the more important naval qualities, it should be subordinated to them: as economy in fuel is advantageous to both, it is all the more important.

Of two equally efficient methods of propulsion, the direct drive is unquestionably preferable. While the transmission of large powers through gearing is not attractive mechanically, it is preferable because of its decidedly better economy.

The tests of the experimental gear at East Pittsburg demonstrated beyond doubt the ability of the reduction gear to transmit large powers with high efficiency, but its suitability for naval vessels depends on its satisfactory operation in actual service, and not until the latter has been proved conclusively can it be pronounced suitable.

One quality of importance that remains to be demonstrated is its durability. Gear experience and the tests of the experimental gear indicate that the life of the teeth under service conditions will be from four to six years. Even if it were only three, the durability would be quite satisfactory. (Note. —This refers only to the wearing parts that will require renewal, and particularly to the pinion.)

The builders are quite confident that its durability—in fact, that its entire operation—will be quite satisfactory from the beginning.

If these expectations are realized—there is good reason to believe they will be—the success of the reduction-gear machinery will be assured, and there will be set a new standard of economy.

RESULT OF TESTS OF A TYPE OF MOSHER WATER-TUBE BOILER BUILT FOR U. S. S. BATTLESHIPS *KEARSARGE* AND *KENTUCKY*, WITH DESCRIPTION OF DESIGN AND NOTES ON CONSTRUCTION.

By Captain R. T. Hall, U. S. N., Member.

The battleships *Kearsarge* and *Kentucky*, built by the Newport News Shipbuilding Company and fitted with Scotch boilers, are undergoing extensive overhauling and repairs at the Philadelphia and Norfolk Navy Yards, respectively, which includes complete replacement of boiler installations. Eight water-tube boilers of the Mosher type are about to be fitted in lieu of the three double-ended and two single-ended Scotch boilers originally installed on each vessel, which latter have been in service since first commissioning of these battleships in 1900.

Contract for new boilers was awarded to the Mosher Water Tube Boiler Company, Ossining, N. Y., the lowest bidder, at a price of $9,741.20 for each boiler. The following general stipulations were contained in the contract:

(*a*) All pressure parts to be constructed entirely of open-hearth steel plates, seamless-steel tubes and seamless headers, if used.

(*b*) No screw joints to be in contact with fire.

(*c*) Generating tubes to be straight or curved, and not less than two inches in diameter.

(*d*) Proper provision to be made for removing and renewing tubes.

(*e*) Method of baffling to be arranged for proper circulation of products of combustion, and facilities provided for readily cleaning soot off tubes and baffles.

(*f*) Boilers to be be designed for a steam working pressure of 180 pounds per square inch; sections, boxes, steam and water drums and all other parts ubject to steam pressure to be tested and made tight under a hydrostatic

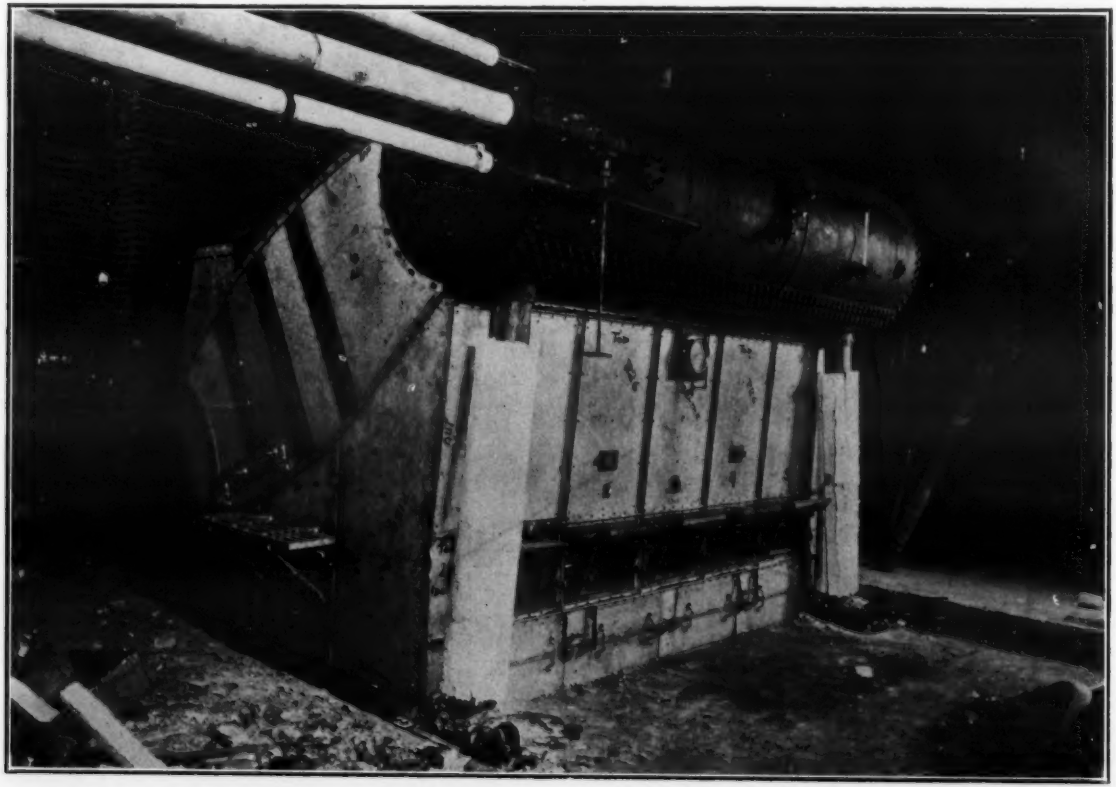


Plate I.—Mosher Type of Boiler Assembled Ready for Test.

pressure of 500 pounds per square inch at the works of the builders. After erection in the vessels all boilers to be subjected to a hydrostatic test pressure of 270 pounds per square inch, and they must prove absolutely tight under this pressure.

(*g*) Boilers to be delivered in such sections as will insure their passing through hatch openings in the deck, the boilers to be afterwards assembled in place on board ship.

A longitudinal sectional view of the Mosher boiler is shown on Plate IV. In order to carry out tests on this type of boiler and ascertain its evaporative efficiency, one boiler was assembled complete at the works of the builders. Photographic illustrations, Plates I, II and III, show this test boiler at various stages of construction.

Specifications, prepared by the Navy Department for these boilers, contained stipulations as follows:

There will be required specific guarantee of the evaporative efficiencies of the boilers proposed when burning coal at the rates of 15, 25, 35 and 40 pounds per square foot of grate surface per hour. The fulfillment of the guaranteed efficiencies to be ascertained by subjecting the first boiler completed, and before it is installed in the vessel, to a series of evaporative tests under both natural and forced draft. Each test to be of at least 24 hours' duration and made under the supervision of representatives of the Bureau of Steam Engineering, Navy Department, and to be at the expense of the contractors.

The tests will be continuous, each to be of 24 hours' duration, with an interval of two hours between each two tests for the purpose of cleaning fires. When forced draft is in use the maximum air pressure must not exceed an equivalent of two inches of water.

During the tests the only cleaning of the boiler tubes that will be allowed will be by the use of steam or air lances, the same as would be the case in actual service. The tests will begin at the lowest rate of combustion called for and will be made in the order stated, the last test being at the highest rate of combustion required. The fuel used shall be clean bituminous coal and may be hand-picked if desired by the contractors.

At the highest rate of combustion the rate of evaporation from and at 212 degrees F. must not be less than 11 pounds of water into dry steam per pound of combustible. Each trial will be at the maximum specified steam pressure, and the maximum specified air pressure must not be exceeded at any time.

Should the evaporation at the highest rate of combustion fall short of the specified 11 pounds of dry steam per pound of combustible by 5 per cent., the type of boiler submitted shall be rejected. A penalty at the rate of $15,000 per every 1 per cent. below the amount of evaporation specified will

be forfeited by the contractors unless the deficiency should amount to 5 per cent., when, as before stated, the type of boiler tested shall be rejected.

Should the type of boilers be rejected on account of failure in the above tests, they will be replaced at the cost of the contractors with boilers of another type, which will fulfill the above guaranteed requirements, or the contract may be canceled, at the option of the Navy Department.

DESCRIPTION OF BOILER.

Referring to Plates I–IV, it will be seen that the Mosher water-tube marine boiler consists of a cylindrical steam drum and oval semicircular water drum connected by large return pipes, 8 inches internal diameter, at each end, which serve to carry the weight of the drums and to provide for water circulation. The steam and water drums are further connected by a series of slightly-curved boiler tubes which stand at a mean angle of 31 degrees to the horizontal. Feed water enters the upper drum. The ends of the 8-inch pipes are connected by riveted saddle flanges to the drums, and at the bottom to two horizontal return pipes 10 inches in diameter. The whole is encased in fire brick, asbestos millboard and galvanized steel-plate casing, which is carried by a substantial frame formed of steel channels and angles tied together by gusset plates. The grates are supported by bearer bars at front, middle and back of furnace, which in turn are supported by brackets and pedestals. The furnace contains 91¼ square feet of grate area; the height of combustion chamber above the grate is 5 feet 8 inches at the front and 2 feet at the back. The total volume of the furnace and combustion chamber is 349.5 cubic feet. The inclination of the grates towards the back is 10 inches in seven feet, and each furnace is provided with five balanced automatic inswinging fire doors, 12 inches high by 18 inches long. Ashpits are 28 inches deep at the front and 18 inches at the back end of the grate, fitted with four portable doors 20 inches by 32 inches at the front, and two hinged doors, 31 inches by 8 inches, at the back, which latter are operated by levers extending to the front of the boiler.

Each boiler weighs about 23 pounds per square foot of heating surface with water 68 degrees F. at steaming level.

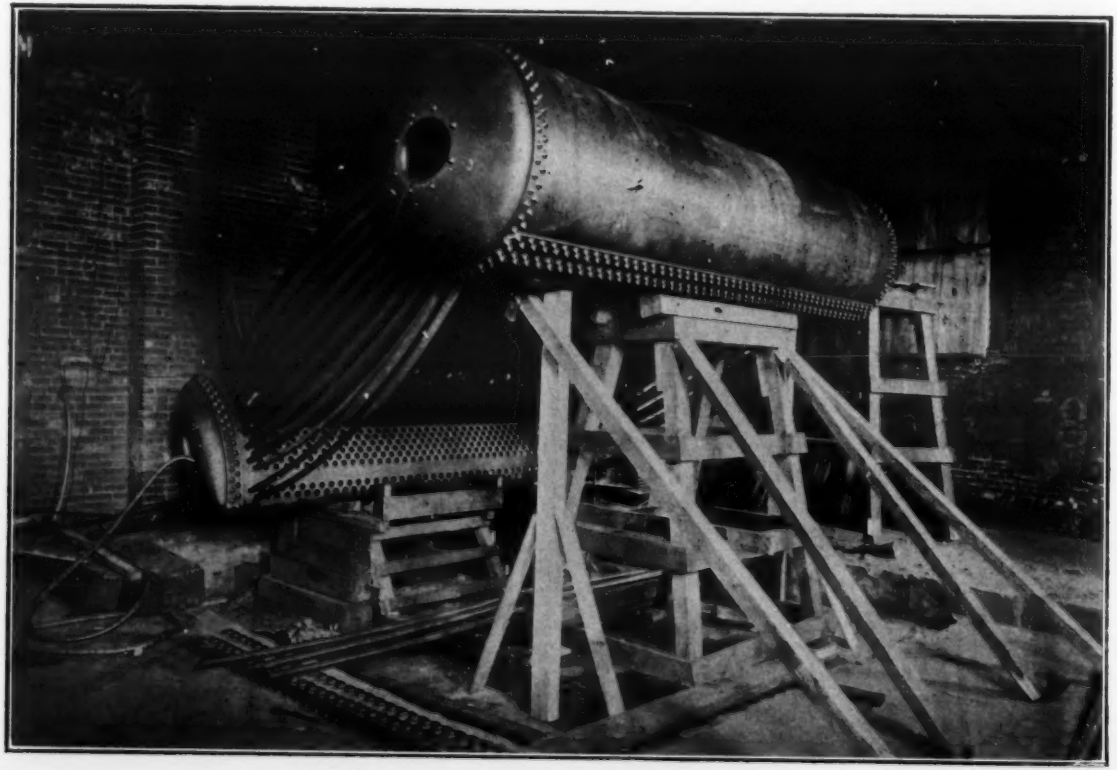


Plate II.—Mosher Boiler in Process of Erection.

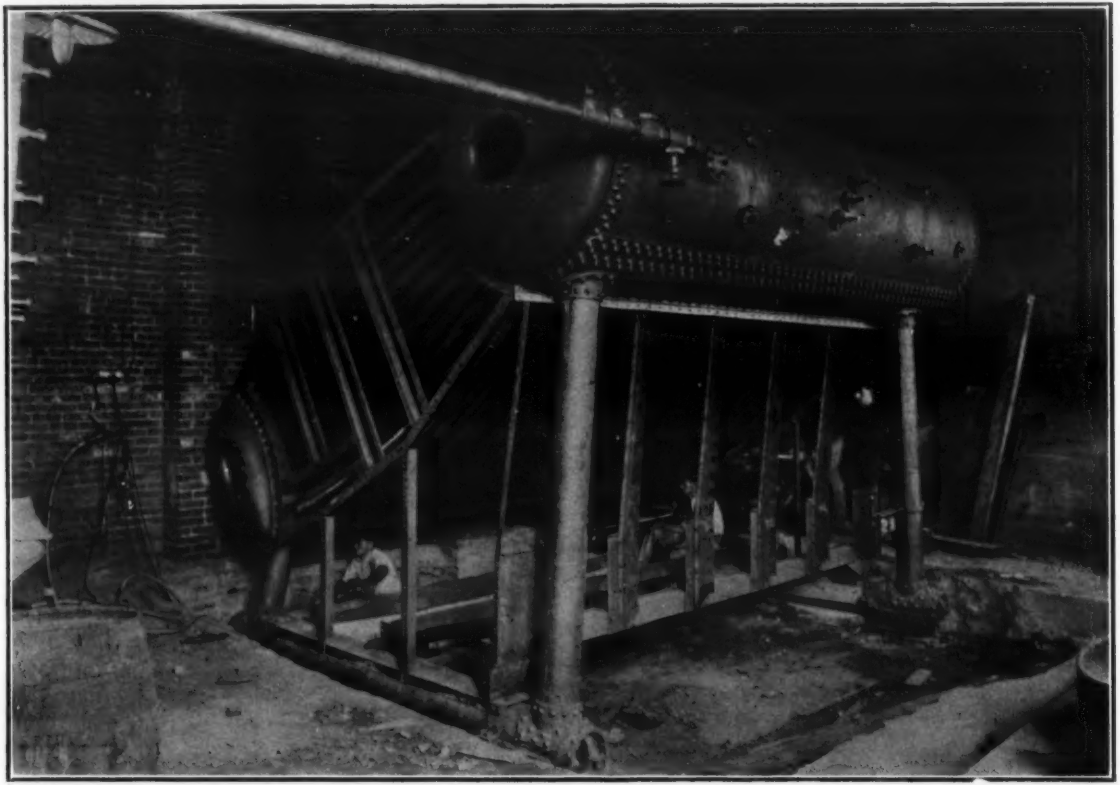
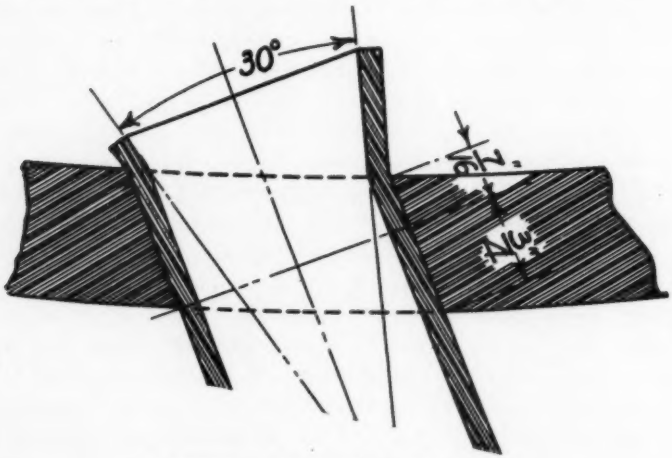


Plate III.—Mosher Boiler in Process of Erection After Tubes have been Expanded and with Vertical Down-Comers and Horizontal Return Pipes in Place.

Steam Drum.—The steam drum is made of one shell sheet, 14 feet 7 inches long by 7 feet 1¼ inches wide and ⅝ inch thick, and one tube sheet 14 feet 7 inches long by 5 feet 9 inches wide, 1⅛ inches thick to compensate for loss of strength in plates due to cutting tube holes. The sheets are butt-jointed with double longitudinal butt straps ½ inch thick, secured by four rows of ⅞-inch rivets in a 4¾-inch lap, giving 90.5 per cent. efficiency of joint. The drum heads are pressed and machined from sheets of steel, and secured to shell sheets by two rows of 1 5/16-inch rivets in a 4½-inch lap, giving 67.35 per cent. efficiency of joint. In each drum is an oval manhole 11 inches by 15 inches.

In the manufacture the shell and tube sheets are laid out and the tube sheet marked for planing. The tube sheet is then planed for the head seams to ⅝-inch thickness and beveled for calking. The sheets are next pressed to correct form by aid of a template curved to 24-inch radius in a powerful vertical hydraulic press. After forming, the sheets are planed parallel on the longitudinal edges and the butt-straps planed, beveled for calking and then pressed to form in the same manner as the shell sheets. A sufficient number of holes are drilled in the butt straps, drum sheets and heads to permit bolting the different parts securely together, the parts being held in position by neat-fitting bolts. The rivet holes are then drilled, and when the drilling is completed the sheets are separated and burrs removed; the drum is then again assembled and the longitudinal seams riveted by a hydraulic stationary riveter. Drum heads are riveted in place by pneumatic riveters. Openings for fitting pads are next cut and pads secured. All seams are then calked and blank flanges fitted to pads by studs, the manhole plates put in place and the drum subjected to a hydrostatic test of 500 pounds per square inch. Next, position of the tube holes is laid out upon the tube sheets. There are 19 rows of tubes, having 46 and 47 tubes, respectively, in each alternate row, and the tubes are triangularly spaced 2¾ inches apart. The drum is secured on a special carriage, under a multiple-spindle

drill press carrying six drills, so that six holes are drilled simultaneously. All but the two middle rows of tubes enter holes drilled more or less tangent to the circumference of the drums, and the opposed bearing surface in tube sheet varies from $\frac{7}{16}$ inch to the full thickness of tube sheet ($1\frac{1}{8}$ inches for steam drum and 1 inch for water drum), as shown in the sketch below. The tangent drilling is done with the aid of a jig or drill guide, which retains the drill in proper position.



SECTION OF TUBE AND DRUM SHEET.

The lower end of each guide is further stiffened by two pointed set screws which are set down on the drum. Pilot holes, $1\frac{1}{4}$ inches diameter, are first drilled, and afterwards the finished holes, which are $2\frac{1}{16}$ inches diameter in the steam drum and $2\frac{1}{32}$ inches diameter in water drum. The steam drum is finally taken from the press and burrs and sharp edges of tube holes removed both on the inside and outside.

Water Drum.—The water drum is constructed similar to the steam drum, except that the longitudinal seams are lapped. The tube sheet is formed to a radius of 30 inches and its sides are flanged to form the inside lap and scarfed on the ends where the shell sheet laps over it. The corners of

the sheets are heated by gasoline torch and the drum heads and tube sheets formed, while hot, to a close fit. The side seams are double riveted with $\frac{15}{16}$-inch rivets in a $4\frac{3}{4}$-inch lap, giving 70 per cent. efficiency of joint. All drum sheets are of steel, tensile strength 58,000 to 65,000 pounds and elastic limit 29,000 to 32,000 pounds. Pads for valve and pipe connections are of cast steel.

Tubes.—Except those in the center row, which are straight, the tubes are bent to nine different curves. They are so curved that they point to a handhole in the opposite side of the steam drum, through which they may be removed and replaced without disturbing other tubes. These holes, however, are omitted in the boilers for the *Kearsarge* and *Kentucky*, and the tubing is done from the furnace.

Tubes are bent to form in a special roller-bending machine and expanded by means of pneumatic-tube rollers. They are flared at the ends by a conical mandrel, and after flaring they are re-rolled to insure tight joints.

Vertical Down-comers and Horizontal Return Pipes.—The vertical down-comers are cold drawn, 8 inches in diameter, of seamless steel, 7 feet 3 inches and 2 feet 2 inches in length, respectively. They are hand riveted to saddle flanges of drums and horizontal return pipes. The latter are hot-drawn seamless steel, 10 inches in diameter (8 inches in diameter being the largest cold-drawn pipes at present manufactured), 11 feet $5\frac{1}{2}$ inches long and $\frac{3}{8}$ inch in thickness. In either end of the horizontal return pipes a drop-forged steel head is riveted, in which an oval handhole, 4 inches $\times$ 6 inches, is fitted.

Baffles.—For collecting soot there is a space 4 inches wide between the back ends of the two lower rows of tubes and the tubes above. Cleaning doors in the casing are opposite this space on either side of the boiler, which open to a semi-horizontal baffle of fire tile laid on the tubes next to the lower row. This baffle extends from the water drum at the back end of the furnace towards the steam drum about two-thirds of the length of the tubes. At the upper end of this semi-

horizontal baffle, and at right angles to it, there is erected a semi-vertical baffle extending about half way up and across the group of tubes. On top of the upper tubes, extending from the steam drum two-thirds of the length of the tubes towards the water drum, is fitted a semi-horizontal baffle forming the roof of the boiler. From the end of this roof, and at right angles to it, a second semi-vertical baffle half way across the tubes is fitted, thus causing the gases, after leaving the furnace, to pass three times across the tubes before reaching the uptake.

The upper semi-horizontal baffle consists of 4 inches of fire tile, 2 inches of asbestos fire felt, $\frac{1}{4}$ inch asbestos millboard and No. 14 U. S. G. steel plate. The lower baffle consists of 2-inch fire tile covered with $\frac{1}{4}$-inch asbestos millboard. The semi-vertical baffles are made of steel strips, $\frac{3}{8}$ inch by $2\frac{3}{4}$ inches wide, 13 feet 7 inches long, neatly fitted between the tubes in double thickness with staggered joints.

Furnace Lining.—The furnace is lined on the front, back and sides with fire brick, $4\frac{1}{2}$ inches, 9 inches and 6 inches thick, respectively, secured to the casing by pocketed bolts, the heads of which are covered with a thick coating of fire clay in the pockets to protect them from the hot gases.

Between the plating covering the front of the furnace and the casing there is an air space connecting with the ash pan through which air is heated on its upward passage and is admitted through perforated brick into the furnace above the fuel. The brick work surrounding the furnace and the tubes is insulated with 2-inch asbestos fire felt enclosed in the boiler casing. The boiler is provided with five inswinging doors, which are so balanced as to close automatically in event of a tube bursting.

Bearing Bars, Foundations, Casing, Etc.—Grate bars are supported at the front, middle and rear by steel bars $\frac{3}{4}$ inch $\times$ $4\frac{1}{2}$ inches $\times$ 13 feet $11\frac{1}{4}$ inches. The middle bearer consists of two bars, of the dimensions noted, bolted together by six $\frac{5}{8}$-inch bolts, which pass through $\frac{3}{4}$-inch pipe distance pieces $3\frac{1}{2}$ inches long.

The foundations consist of galvanized-steel angles and channels which support movable galvanized-steel ash pans, four to each boiler.

The boiler casing, which is tightly fitted in sections, is of galvanized-steel sheets, attached to angles. On the sides doors are provided to permit cleaning with steam lances and for the removal of soot. The casing lining is of 2-inch magnesia blocks and 1¼-inch asbestos millboard.

BOILER DATA.

Length of boiler, feet and inches	14–09⅝
Height of boiler, feet and inches	12–09
Width of boiler, feet and inches	15–02
Weight of boiler, including water to steaming level, pounds	91,551
Steam Drum:	
Length, feet and inches	14–08
Inside diameter, inches	0–48
Tube sheet, thickness, inches	0–01½
Shell sheet, thickness, inch	0–00⅝
Heads, thickness, inch	0–00⅝
radius of curvature, inches	0–48
Size of manhole, inches	11 × 15
Water Drum:	
Radius of curvature, semi-circular, inches	0–48
Greater diameter, inches	0–44
Lesser diameter, inches	0–24
Tube sheet, thickness, inch	0–01
Shell sheet, thickness, inch	0–00 9/16
Heads, thickness, inch	0–00 9/16
radius of curvature, inches	0–48
Size of manhole, inches	11 × 15
Tubes, Heating and Grate Surface, etc:	
Number of tubes (seamless, cold-drawn)	883
Tubes, outside diameter, inches	0–02
thickness, No. B.W.G	8
length of longest tube, feet and inches	9–06⅝
shortest tube, feet and inches	8–02¾
Heating surface, square feet	3,980.3
Water-heating surface, per cent	98.3
Superheating surface, per cent	1.7
Grate, length, feet and inches	7–00½
width, feet	13–00
surface, square feet	91.25
Ratio of heating surface to grate	43.6 to 1

Tubes, Heating and Grate Surface, etc.:

Furnace, height at front, feet and inches	5–08
back, inches	0–23
Boiler, external length, feet and inches	13–03
width, feet	15–00
height, feet and inches	12–09
Grate bars, kind	Navy standard.
number of	140
Air space, inch	0–00½
Ratio of grate to air	9 to 8
Fire doors, number of	5
Normal horsepower	1,500
Designed working pressure, pounds per square inch	180

The following fittings are attached to one boiler:

1 6-inch steam stop valve.
4 swash plates.
1 dry pipe.
1 zinc basket.
1 2½-inch main-feed stop valve.
1 2½-inch main-feed check valve.
1 2½-inch main-feed internal pipe.
1 2½-inch auxiliary-feed stop valve.
1 2½-inch auxiliary-feed check valve.
1 2½-inch auxiliary-feed internal pipe.
2 1½-inch blow valve with internal pipe.
1 scum pan.
1 brass-body steam gage.
3 gage cocks.
2 water gages fitted with Dewrance automatic closing device.
2 drain cocks.
1 air cock.
1 ¾-inch stop valve for cleaning pipe.
1 connection for testing water.
1 cleaning hose and lance.

A system of pipes in the passages is provided with steam and compressed-air connections for blowing soot from the boiler tubes.

RESULTS OF TESTS, WITH EXTRACTS FROM THE BOARD'S REPORT.

In accordance with the terms of the contract a Naval Board composed of Commander C. W. Dyson, Lieutenant Commander J. K. Robison and Lieutenant Diman, was appointed by the Navy Department to carry out economy tests on this type of boiler at the works of the builders. Ten naval officers, in addition, were detailed to assist the Board in collection and compilation of data.

Table A contains a summary of important particulars of the tests, which are also graphically illustrated on Plate VI.

Curves showing analysis of smokepipe gases are given on Plates VII–X.

General Arrangement of the Testing Plant.—The boiler was erected in a brick building, the entrance door being fitted with an air lock for use under forced draft, and windows arranged so they could be secured airtight.

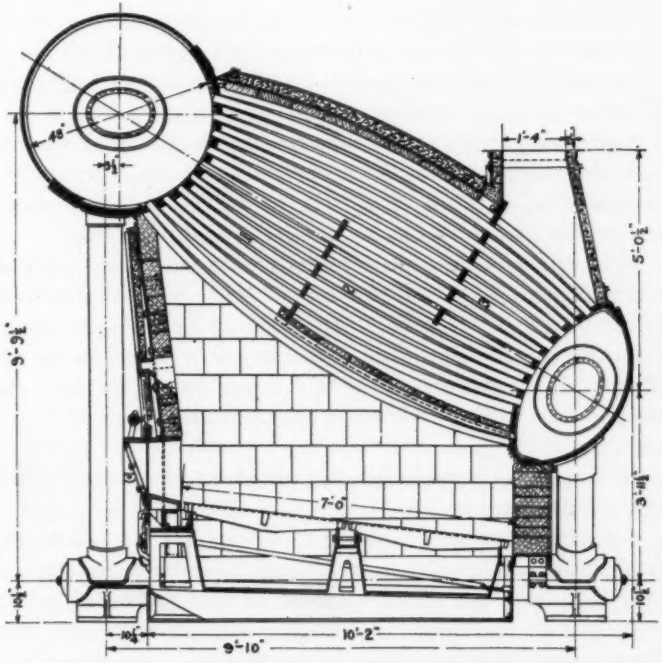


Plate IV.—SECTIONAL VIEW OF MOSHER WATER-TUBE BOILER FOR U. S. S. "KEARSARGE" AND "KENTUCKY."

ONE BOILER.

Total tube heating surface, square feet	3,918.66
Total water drum heating surface, square feet	34.62
Total steam drum heating surface, square feet	27.02
Total heating surface, square feet	3,980.30
Area of grate surface, square feet	91.25
Ratio of heating surface to grate surface	43.6 to 1

The main steam pipe from the steam drum rose vertically, passing through the overhead floor to a height of about seven feet from the drum, where were fitted a Navy standard steam gage, a throttling calorimeter, safety valves and a bleeder valve, by means of which latter an even pressure of steam could be maintained. There was also a connection to the main steam line of the works.

Forced Draft.—The necessary air for combustion, at other than natural draft, was supplied by a large Buffalo Forge Co.'s fan, driven through a belt and flywheel by a single-cylinder, vertical, inverted, direct-acting steam engine.

The blower was located directly outside the boiler house and discharged air into the fireroom diagonally towards the back of the boiler, thus causing a current of air to sweep around the fireroom from back to front of the boiler.

Feed Heater.—A multi-coil Reilly feed heater was used for maintaining a high temperature of feed water, the heating medium being live steam. The heater, which was proven for tightness at the beginning and conclusion of the tests, was fitted with a continuous recording temperature gage, and the temperature during trials was held practically constant during each test.

Steam Jet.—A steam jet was fitted in the smoke stack beyond the damper.

Uptakes, Flue and Smokestack.—The gases of combustion, after leaving the boiler, rose through a vertical uptake to a long horizontal flue which extended to the smoke stack on the outside of the building. A swinging damper, controlled from the fireroom floor, was fitted in this flue.

Instruments for Pressures and Temperatures.—Calibrated thermometers were placed in the feed line on the discharge side of the heater, close to the boiler, and in the suction line to feed pump from measuring tank. The temperature of the fireroom was taken by means of a thermometer placed under the boiler grating at the front of the boiler in order to ascertain the probable temperature under similar fireroom conditions on board ship. A thermometer was placed outside of

the test room to secure temperatures of the outside air. A continuous recording thermometer was placed on discharge side of heater, and a continuous recording pressure gage on main steam line. A Taylor Cambridge electric pyrometer was installed for taking the temperature existing in the different passes of the boiler, and an ordinary nitrogen pyrometer fitted in the flue to obtain temperature of the escaping gases. An aneroid barometer was hung outside of the test room.

Draft pressures were taken by a tube inserted through dusting doors in the different passes, and the air pressure in the fireroom was measured by the same differential gage. A U-tube was located in the smoke stack to obtain the draft.

Gas Analysis.—An Orsatt apparatus for making analyses of the furnace gases was located in the fireroom. Gas samples were taken continuously from the uptake by means of a pipe leading to the Orsatt apparatus. Samples of gas were taken at frequent intervals, and from results of these analyses curves have been plotted, as shown on Plates VII, VIII, IX and X. It should be noted that curves show average conditions and not instantaneous ones, as the gas was drawn continuously and not intermittently.

Method of Weighing Coal and Ashes.—A platform scales for weighing coal was placed outside the fireroom. Fuel, weighed in barrows as needed, was a mixture of two varieties, viz: Somerset and Pocahontas coal. At the end of each hour the fireroom floor was cleaned up and an estimate of coal remaining on the floor made from which the total burned for the hour was figured.

The ash was removed at intervals as found necessary; at the end of each test ash pans were thoroughly cleaned and the net weight figured for the test.

Method of Weighing and Pumping Water.—The feed water was delivered from the city mains into two cylindrical tanks placed on separate scales. Each tank was fitted with a quick-opening valve and discharged into a feed tank, below which was calibrated and fitted with a gage glass so that the water could be checked up at any time. The full and empty

tanks were weighed and net weight of water dumped into the feed tank was recorded. The feed pump took its suction from the feed tank and discharged through the heater into the boiler. The speed of the pump was controlled by an attendant stationed at the heater, who kept the temperature of the feed, and height of water in the boiler, as nearly constant as possible. The height of water in the feed tank and also in the boiler was noted at the beginning and end of each test, so that the total amount of water fed to the boiler could be accurately ascertained.

Coal.—The coal burned was hand-picked Pocahontas and ordinary run-of-mine Somerset. These were mixed, as there was considerable difficulty burning the Pocahontas alone on account of clinkers. The coal used was taken from the same lots during all tests. The proportion of the mixture was changed during the tests, as follows: First test, two parts by weight of Pocahontas to one of Somerset; second test, five parts Pocahontas to two of Somerset; third test, five parts Pocahontas to one and one-half of Somerset; on the fourth test the same mixture as on the third was used until 4 A. M., when for a period of four hours Pocahontas only was burned. The coal was only partially under cover, and therefore much of it exposed to weather conditions.

Method of Firing.—The operation of the boiler and of the firing was under direction of Consulting Engineer G. H. Barrus, of Boston, Mass., who supervised the tests for the Mosher Boiler Co. A schedule of firing was laid out and closely adhered to, as follows:

On the 15-pound test, regular round of two firings and one leveling at five-minute intervals were made, or eight firings per hour—doors 1, 2 and 3, on every five minutes, and doors 4, 5, two minutes later. Three small shovelfuls were put on at each firing. Fires were sliced every two hours and cleaned during the last hour of test.

On the 25-pound test, the same method of firing was carried out as on the previous test, except four to five small shovelfuls were used at each firing. Slicing was done every two hours and fires cleaned between 4 and 5 P. M. and at 7 A. M.

On the 35-pound test, regular rounds of firings with five-minute intervals were made, with leveling three minutes after every second firing, making twelve firings and six levelings per hour. Four small shovelfuls were put on at each firing. Sliced every two hours and cleaned fires every four hours.

On the 40-pound test the same firing was carried out as on the previous test, except four to five shovelfuls were used in each door at each firing. Sliced every two hours and cleaned fires every four hours.

Firemen Employed.—The firemen employed, with few exceptions, were of the ordinary type secured through agencies in New York. They worked in three shifts of four hours' duration.

Methods of Starting and Stopping Tests.—Before the first test the fires were cleaned, ash pans hauled and fires brought to a uniform estimated thickness of four inches. As no time was desired between tests by the company, the fires being cleaned on the last hour of each test, the end of one test was the beginning of the next. The depth of fires was estimated at the beginning and end of each test, and, as will be seen from the following, was nearly the same for all tests:

Beginning of test, No. 1, inches	4
End of test, No, 1, beginning of No. 2	4½
No. 2, beginning of No. 3	5
No. 3, beginning of No. 4	5
No. 4	5

Moisture in Coal and Ash.—Samples of coal and ash, which were taken during each test, were preserved in airtight jars, and later the moisture determined by laboratory test.

Water was used at times in the ash pans, but generally only when fires were cleaned. The net weight of ash for each test was corrected in accordance with the amount of contained moisture.

Quality of Steam.—A throttling calorimeter was fitted in the main steam pipe at a point about 7 feet above the steam

drum. The collecting nozzle was of standard pattern. The calorimeter was calibrated after the tests by taking readings with the pressure both rising and falling with little or no steam leaving the boiler except through the calorimeter. From these standard readings the amount of moisture was calculated by the formula $Q = \frac{.48(T-t)}{L} \times 100$, in which Q = per cent. moisture, T = calibration reading of lower thermometer, t = test reading of lower thermometer, L = latent heat of the steam at the boiler pressure.

NOTES ON THE TESTS.

Before the tests were undertaken the boiler was opened for examination and everything found in good condition. The boiler was then closed and subjected to a hydrostatic pressure of 180 pounds per gage. No leaks were detected. The water in the boiler was next run down to steaming level and all blow joints broken. The feed heater was proven before and after the tests to 200 pounds' water pressure per gage and found tight.

Four test runs were made, as required by the contract, at the following rates of combustion: 15, 25, 35 and 40 pounds of coal per square foot of grate per hour, each test being of twenty-four hours' duration.

Test No. 1, at 15 Pounds of Coal per Square Foot of Grate per Hour.—This was begun at 8 A. M., November 1, and finished at 8 A. M., November 2, 1910. The weather during the test was clear.

The blower was not run, the fireroom compartment was open and the steam jet in stack was partly open.

The rate of evaporation from and at 212 degrees F. per pound of combustible was 12.09 pounds.

Test No. 2, at 25 Pounds of Coal per Square Foot of Grate per Hour.—This test began at 8 A. M., November 2, 1910, and ended at 8 A. M., November 3, 1910. The weather was clear at the beginning of test, cloudy at about 4 P. M., rain-

ing at 8 P. M., which condition continued until the test ended.

The steam jet in stack was partly open. The forced-draft blower was run slow, giving an average pressure in the fireroom corresponding to 1.04 inches of water.

The rate of evaporation from and at 212 degrees F. per pound of combustible was 11.58 pounds.

Test No. 3, at 35 Pounds of Coal per Square Foot of Grate per Hour.—This test was begun at 8 A. M., November 3, 1910, and ended at 8 A. M., November 4, 1910.

The steam jet in stack was partly open. The forced-draft blower was in operation and the fireroom compartment closed. Air pressure in fireroom averaged 1.65 inches of water.

The weather during this test was rainy. Considerable difficulty was experienced by those in charge in maintaining the fires in good condition.

The rate of evaporation from and at 212 degrees F. per pound of combustible was 10.85 pounds.

Test No. 4, at 40 Pounds of Coal per Square Foot of Grate per Hour.—This test was begun at 8 A. M., November 4, 1910, and ended at 8 A. M., November 5, 1910. The weather was rainy for the first 12 hours, clear during the last 12 hours.

The steam jet in the stack was partly open. The forced-draft blower was in operation and the fireroom compartment closed. An average air pressure of 1.62 inches of water was maintained.

The rate of evaporation from and at 212 degrees F. per pound of combustible was 11.048 pounds, and the actual consumption per square foot of grate 39.91 pounds.

Upon conclusion of the fourth test the boiler was examined; no signs of leaks or distortion of any kind could be observed, and later (November 9), a hydrostatic test of 180 pounds was applied and the boiler found tight. An examination of the interior showed all parts to be in excellent condition. The fire bricks at the rear of the furnace were slightly chipped, due to the use of slice bars, and lining at front and sides of

furnace were slightly blistered. Grate bars were coated with loose clinker, covering about 25 per cent. of the air space.

In preparation for these tests, the Mosher Company carried on preliminary trials of this boiler for approximately one month. The last two weeks of this preliminary testing was under the supervision of the boiler expert, who represented the Mosher Company on the official tests. During this preliminary-test period sufficient firemen were trained to give a good working nucleus for the official tests; also, a very complete schedule of firing for the official tests, as hereinbefore mentioned, was developed. The result of this preliminary work was well shown on the official tests, which were carried out systematically and with great care on the part of the contractors.

Owing to the sharp turns in the uptakes and flue and to the shortness of the smokepipe, a steam jet was used in the flue through all the trials to assist in maintaining the required rates of combustion.

All data were taken simultaneously by representatives of the company and by assistants to the Naval Board, so that all figures were checked at the time of observation.

COMMENTS ON BOILER.

During the first test, when burning coal at the rate of about fifteen pounds per square foot of grate surface without forced draft, the temperature of the working fireroom was rather high, due to excessive radiation from the boiler front. As the boiler is at present built, this appears to be unavoidable, but it may be reduced to some extent by increasing the lagging on the front and fitting such extra thickness over the air space.

Better evaporative results might be obtained with boilers of the type tested if a change were made in the furnace arrangement. By referring to Plate IV, which shows a longitudinal sectional elevation of this type of boiler, it is seen that air entering the furnace through the ashpit or through the fur-

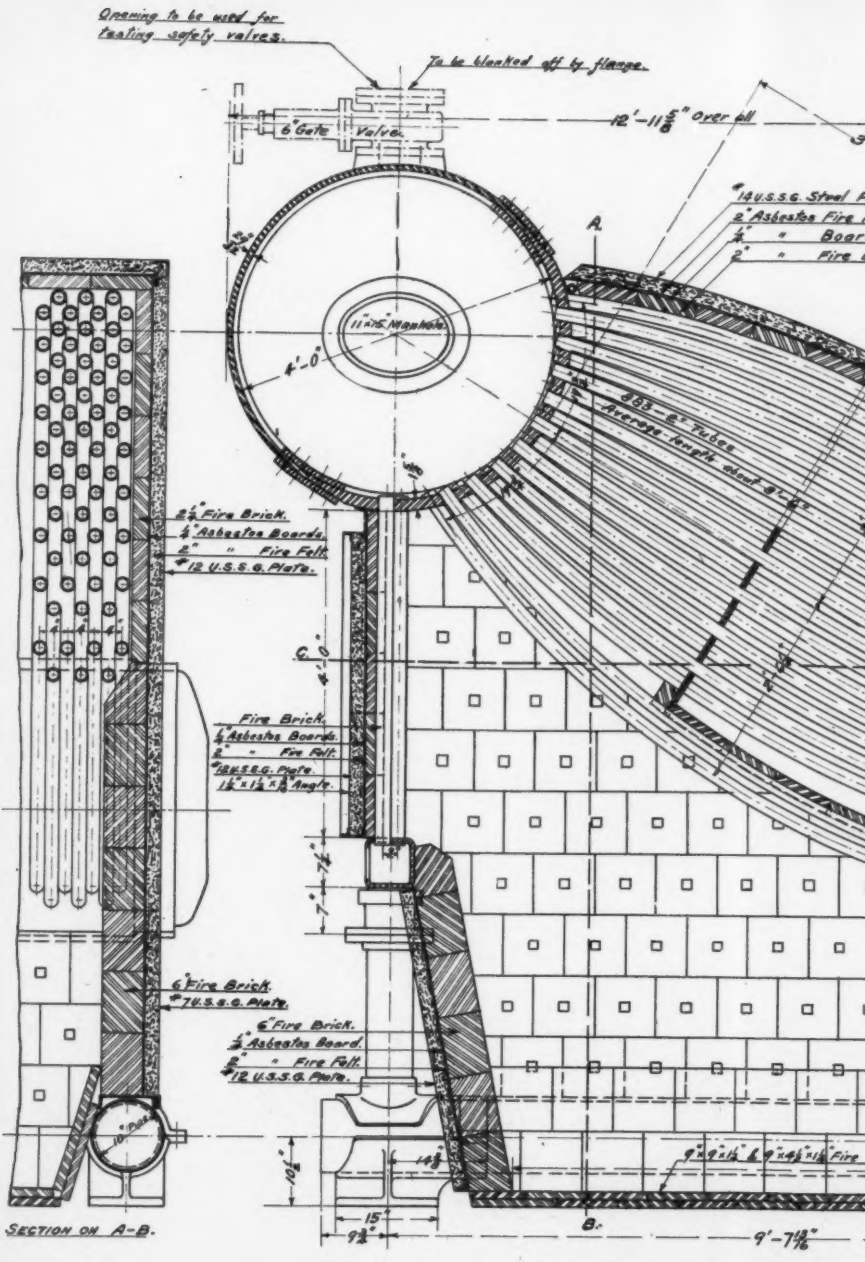
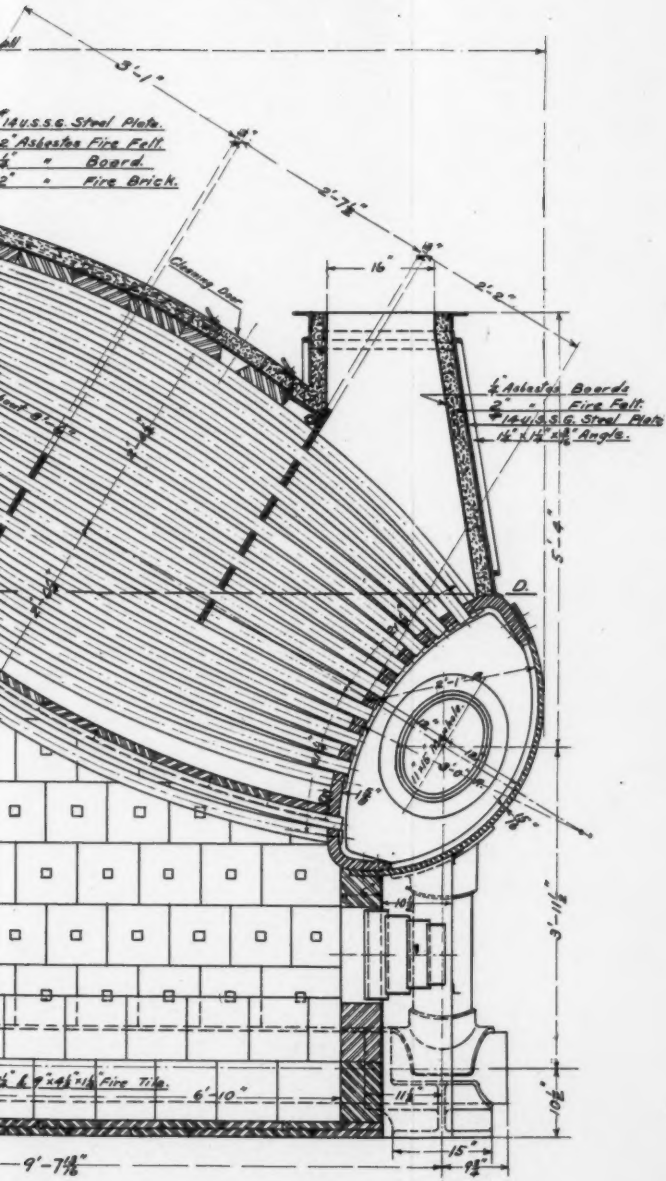


Plate V.—MODIFIED TYPE OF MOSHER BOILER ARRANGE

(Bureau Steam Engineering, Navy Department, Janua



LER ARRANGED FOR OIL BURNING.

Department, January, 1911.)

nace door when opened, passes through the thinnest part of the fire, directly inside the furnace door, or directly through the furnace door, thus entering the first pass without going over the glowing bed of fire on the grates. This results in incomplete mixing of air with the gases passing from the fire, and also a cooling down of the products of combustion just before entering the first draft pass, which condition is believed prejudicial to the highest evaporative efficiency. A change in design of furnace, illustrated on Plate V, is suggested by the Board, which shows a modified type of Mosher boiler with furnace arranged for oil burning.

Table A. NUMBER OF TEST	1	2	3	4
1. Date of test	Nov 1-2, 1910	Nov 2-3, 1910	Nov 3-4, 1910	Nov 4-5, 1910.
2. Duration of test.....hrs.	24	24	24	24
3. Kind of fuel	Pocahontas and Somerset.			
4. Kind of start	Alternate	Alternate	Alternate	Alternate
5. State of weather	Clear	Clear to Cloudy	Rainy	Raining to Clear.
AVERAGE PRESSURES, OR DRAFT.				
6. Barometer.....ins.		30.07		29.71
7. Steam pressure by gauge.....lbs.	178.3	180	180.2	181.2
8. ~~Pressure~~ Draft at ~~base~~ of pipe—ins. of water	-.51	-.51	-.5	-.5
9. ~~Pressure~~ Draft in furnace—ins. of water.....3rd. Pass	—	.47	.084	-.18
10. ~~Force of draft in pounds~~ Pressure in furnace—ins. of water.....2nd. Pass	—	.595	.56	.39
11. ~~Resistance of blower~~.....1st. Pass	—	.758	.95	.78
Pressure in fireroom.	—	1.04	1.65	1.62
AVERAGE TEMPERATURES.				
12. External air.....degrees F.	50	51.5	45	48
13. Fireroom.....do.....	113.5	107.	87.	87.
14. Steam.....do.....	373.5	380.2	380.1	380.9
15. Feed water entering heater.....do.....	—	—	—	—
16. Feed water entering economizer..do.....	—	—	—	—
17. Feed water entering boiler.....do.....	213.0	210.1	204.2	186.1
18. Air entering ash pit.....do.....	113.5	107.	87.	87.
19. Escaping gases from boiler.....do.....	421.	485.	531.	541.
20. Escaping gases from economizer .do.....	—	—	—	—
FUEL.				
21. Kind of	Pocahontas and Somerset.			
22. Weight of wood used in lighting fires.....lbs.	—	—	—	—
23. Weight of coal as fired*.....lbs.	33250	53950	76223	87403
24. Moisture in coal.....per cent.	3.31	2.23	3.31	3.06
25. Weight of dry coal consumed.....lbs.	32132	52744	73700	84754
26. Weight of ash and refuse.....lbs.	2744	3773	4331	4948
27. Weight of combustible consumed...lbs.	29388	48971	69369	79806
28. Per cent of refuse in dry coal.....	8.54	7.15	5.88	5.84
FUEL PER HOUR.				
29. Coal consumed per hour.....lbs.	1385	2248	3176	3642
30. Dry coal consumed per hour.....lbs.	1339	2197	3071	3531
31. Combustible consumed per hour....lbs.	1225	2040	2891	3325
32. Coal consumed per hour per sq. ft. G. S.....lbs.	15.18	24.63	34.87	39.91
33. Dry coal consumed per hour per sq. ft. G. S.....lbs.	14.67	24.07	33.65	38.69
34. Combustible consumed per hour per sq. ft. G. S.....lbs.	13.42	22.35	31.67	36.43

* Including equivalent of wood used in lighting fires.

NUMBER OF TEST	1	2	3	4
35. Coal per hour per sq. ft. H. S. lbs.	.348	.565	.798	.915
36. Dry coal per hour per sq. ft. H. S. lbs.	.336	.552	.771	.887
37. Combustibles per hour per sq. ft. H. S. lbs.	.308	.512	.726	.835
QUALITY OF STEAM.				
38. Per cent of moisture in steam	0	0	0	0
39. Degrees of superheating	—	—		
40. Quality of steam (dry steam = 100)	100	100	100	100
WATER.				
41. Total weight of water fed to boiler* lbs.	339097	539694	712158	818884
42. Water actually evaporated, corrected for quality of steam (40 by 41) lbs.	339097	539694	712158	818884
43. Factor of evaporation	1.0475	1.0507	1.0568	1.0756
44. Equivalent water evaporated into dry steam from and at 212° (42 by 43) lbs.	355204	567056	752609	880792
WATER PER HOUR.				
45. Water evaporated per hour, corrected for quality of steam lbs.	14129	22487	29673	34120
46. Equivalent evaporation from and at 212° lbs.	14800	23627	31358	36699
47. Equivalent evaporation from and at 212° per sq. ft. G. S. lbs.	162	259	344	402
48. Same per sq. ft. of heating surface. lbs.	3.72	5.94	7.88	9.22
ECONOMIC RESULTS.				
49. Water apparently evaporated under actual conditions per lb. of coal as fired (41 ÷ 23) lbs.	10.20	10.00	9.34	9.37
50. Apparent equivalent evaporation from and at 212° per lb. of coal (including moisture) 44 ÷ 23. lbs.	10.68	10.51	9.87	10.08
51. Equivalent evaporation from and at 212° per lb. of dry coal (44 ÷ 25). lbs.	11.05	10.75	10.21	10.39
52. Equivalent evaporation from and at 212° per lb. of combustible (44 ÷ 27). lbs.	12.09	11.58	10.85	11.04
EFFICIENCY.				
53. Efficiency of boiler; heat absorbed by the boiler per lb. of combustible divided by the heat value of one lb. of combustible. (*See* Sheet No. 6)	74.18%	70.97%	66.40%	67.47%
54. Efficiency of boiler, including grate; heat absorbed by the boiler per lb. of dry coal, divided by the heat value of one lb. of dry coal. (*See* Sheet No. 6)	70.66%	68.42%	64.77%	65.60%
REMARKS AND OBSERVATIONS.				
55. Principal data taken every	½ hour	½ hour	½ hour	½ hour
56. Percentage of smoke as observed	From medium to heavy			
57. Method of observing same	—	—	—	—
58. Kind of firing (spreading, alternate, or coking)	Spreading; fires levelled; doors fired in rotation			
59. Average thickness of fires	4" to 4½"	4½" to 5"	5"	5"
60. Average intervals between firings for each furnace during time fires were in normal condition	8 firings, 4 levellings, 3 shovels, 5 min.	8 firings, 4 levellings, 4 to 5 shovels, 5 min.	12 firings, 6 levellings, 4 shovels, 5 min.	12 firings, 6 levellings + 4 to 5 shovels, 5 min.
61. Average interval between times of breaking up	As required			
62. Efficiency of firemen; expert, average, or poor	Average	Average	Average	Average

*Corrected for inequality of water level and steam pressure at beginning and end of test.

FUEL AND GAS ANALYSES.

PROXIMATE ANALYSIS OF FUEL.

	[illegible] Somerset Per cent.	[illegible] Pocahontas Per cent.
Fixed carbon	71.53	75.68
Volatile matter	14.31	13.676
Moisture	4.57	2.344
Ash	8.99	2.30
Total	100.00	100.00
Sulphur separately determined	1.60	.72

ULTIMATE ANALYSIS OF DRY FUEL.

	Somerset Per cent.	Pocahontas Per cent.
Carbon (C)	80.40	88.76
Hydrogen (H)	4.28	4.85
Oxygen (O)	4.03	2.23
Nitrogen (N)	1.30	1.36
Sulphur (S)	1.60	.72
Mineral	8.99	2.03
	100.00	100.00
Moisture in sample of fuel as received		

ANALYSIS OF ASH AND REFUSE.

	1	2	3	4
Combustible	50.55	62.40	64.61	52.42
Moisture	.16	.153	.31	.03
Balance	49.29	37.367	35.08	47.55

CALORIFIC VALUE OF FUEL.

	1	2	3	4	
Kind of calorimeter used					
Calorific value by calorimeter, per pound of dry coal	Sample of P: 15645; S: 14169				B. T. U.
Calorific value by calorimeter, per pound of combustible	P: 15464; S: 15268				Do.
Calorific value by analysis, per pound of dry coal	15173	15239	15297	15370	Do.
Calorific value by analysis, per pound of mixture of combustible	15018	15032	15053	15076	Do.

ANALYSES OF DRY GASES.

	1	2	3	4
Carbon dioxide (CO_2)	12.63	11.63	10.60	11.36
Oxygen (O)	5.93	6.31	7.89	7.23
Carbon monoxide (CO)	.32	.35	.25	.30
Hydrogen and hydrocarbons				
Nitrogen (N) (by difference)	81.02	81.05	82.06	81.09
	100.00	100.00	100.00	100.00

HEAT BALANCE SHEET.—TESTS OF MOSHER W. T. BOILER FOR U. S. S. "KEARSARGE" AND "KENTUCKY."

Number of test	1		2		3		4	
Pounds of dry gas per pound of carbon	19 45		20.86		22.87		21.77	
Heat absorption and losses in B.T.U. or in per cent. of calorific value of combustible	B.T.U.	Per cent.	B.T.U.	Per cent.	B.T.U.	Per cent.	B.T.U.	Per cent.
Heat absorbed by boiler per pound of combustible	11,751.	74.30	11,237.	70.97	10,538.	66.46	10,713.	67.48
Loss due to moisture in coal	40.01	.25	27.14	.17	41.19	.26	38.67	.24
Loss due to burning of hydrogen	449.60	2.85	464.49	2.93	480.87	3.03	492.43	3.10
Loss due to heat carried away in the dry chimney gases	1,752.31	11.08	2,131.78	13.46	2,629.07	16.60	2,504.83	15.77
Loss due to incomplete combustion	783.70	4.96	843.04	5.33	918.56	5.79	869.	5.47
Loss due to unconsumed hydrogen, etc., heating moisture in air, to radiation, etc	1,035.38	6.56	1,128.55	7.14	1,245.31	7.86	1,260.07	7.94
Calorific value per pound of combustible	15,813.	100.00	15,832.	100.00	15,853.	100.00	15,878.	100.00
Efficiency of boiler, per cent	...	74.18	...	70.97	...	66.40	...	67.47
Efficiency of boiler and furnace, per cent	...	70.66	...	68.39	...	64.77	...	65.60

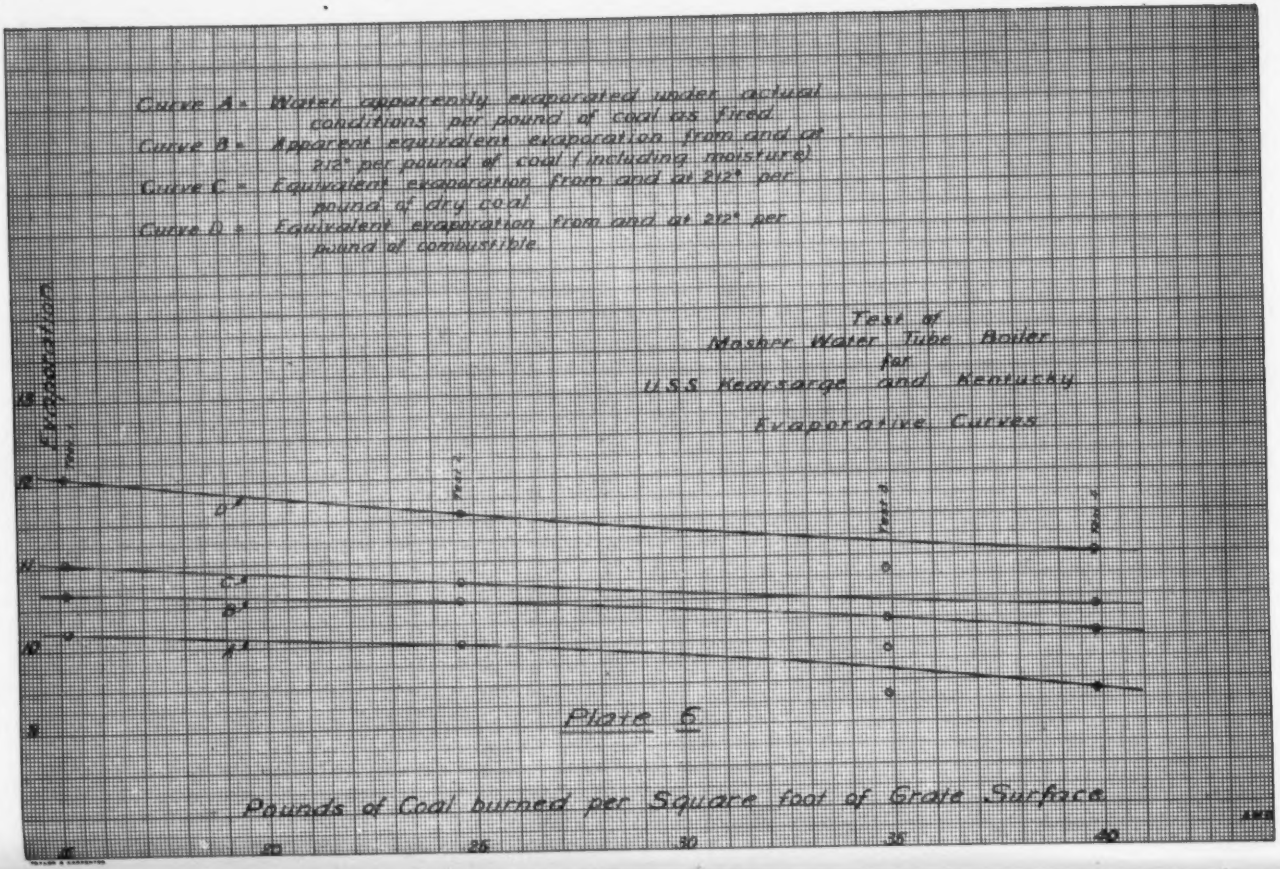

Curve A = Water apparently evaporated under actual conditions per pound of coal as fired
Curve B = Apparent equivalent evaporation from and at 212° per pound of coal (including moisture)
Curve C = Equivalent evaporation from and at 212° per pound of dry coal
Curve D = Equivalent evaporation from and at 212° per pound of combustible
Test of Mashrr Water Tube Boiler for U.S.S. Kearsarge and Kentucky
Evaporative Curves
Evaporation
Test 1
Test 2
Test 3
Test 4
D
C
B
A
Pounds of Coal burned per Square foot of Grate Surface

Plate 6

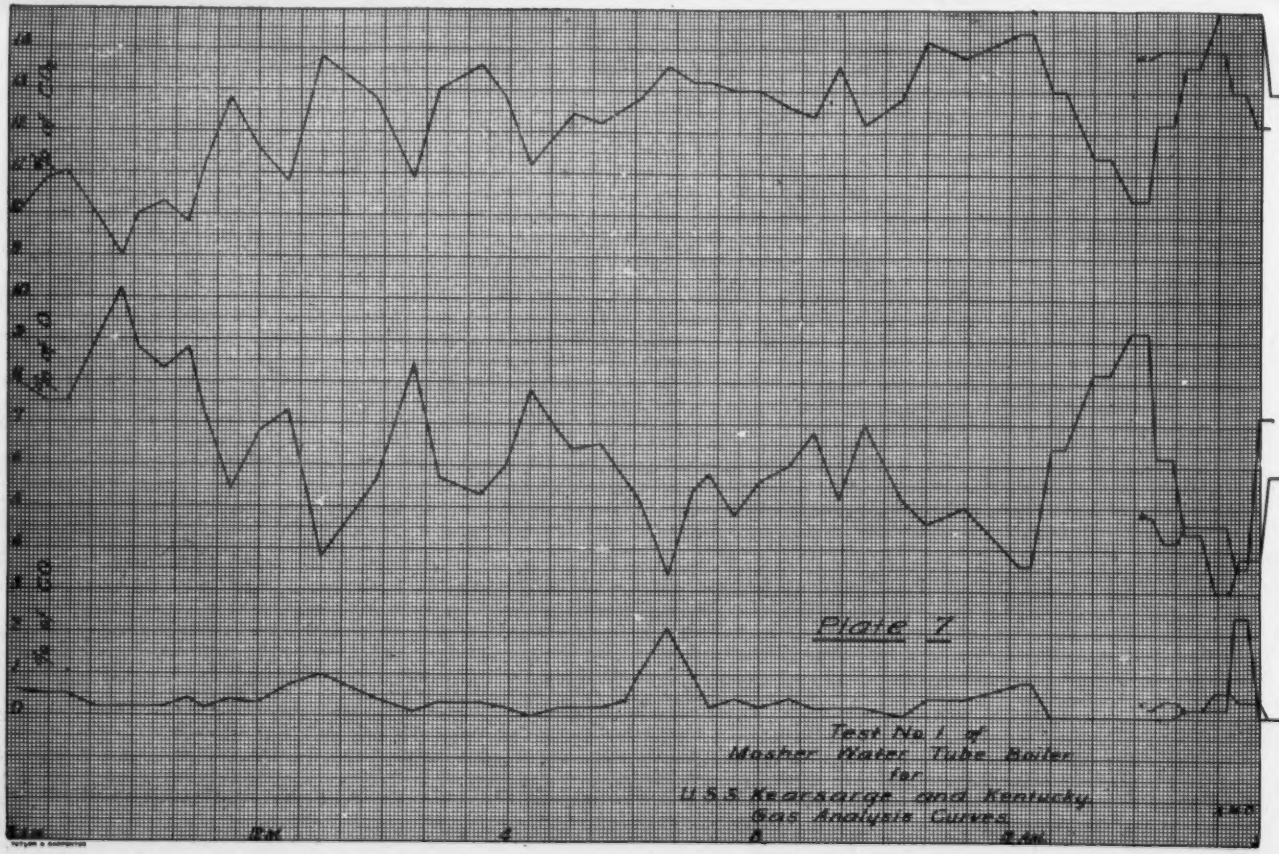
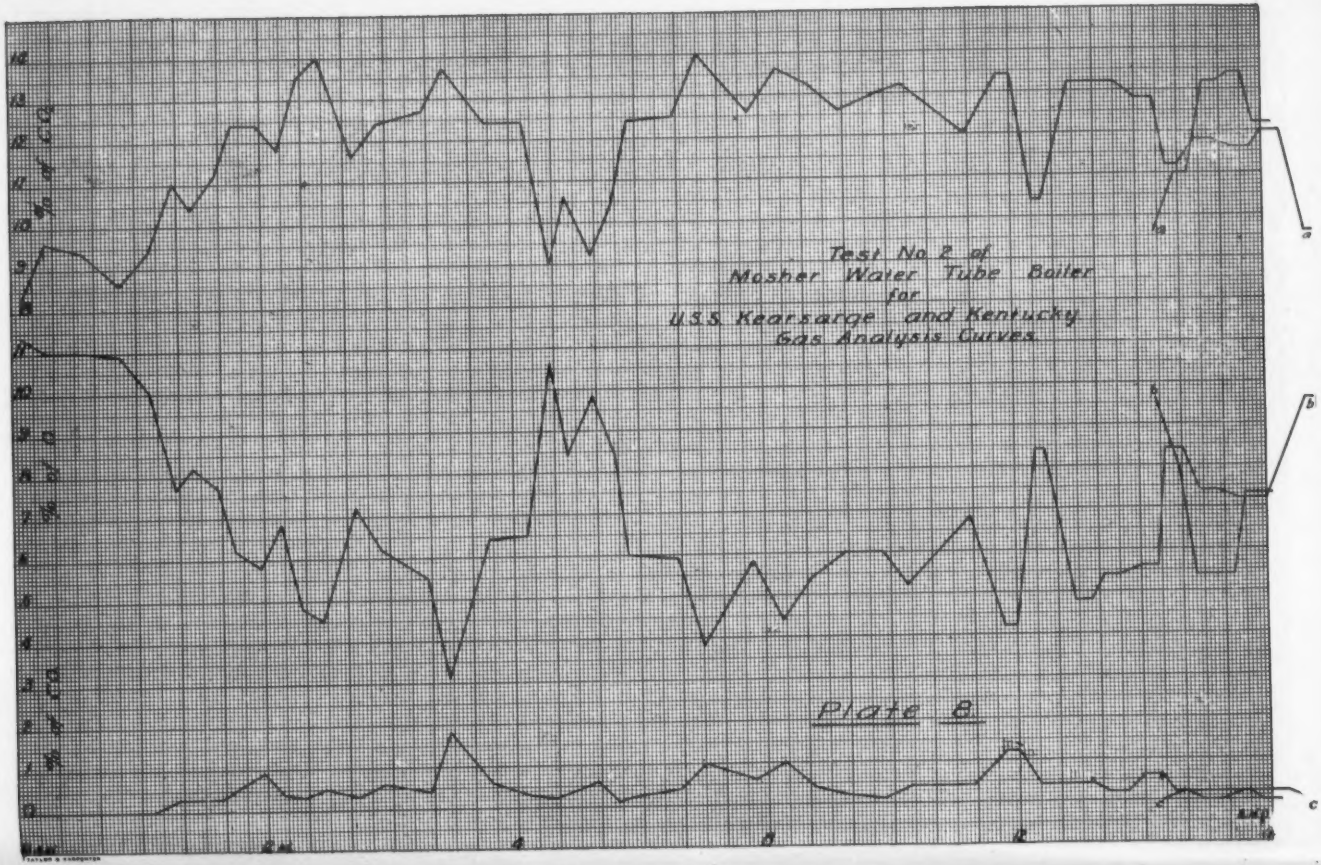
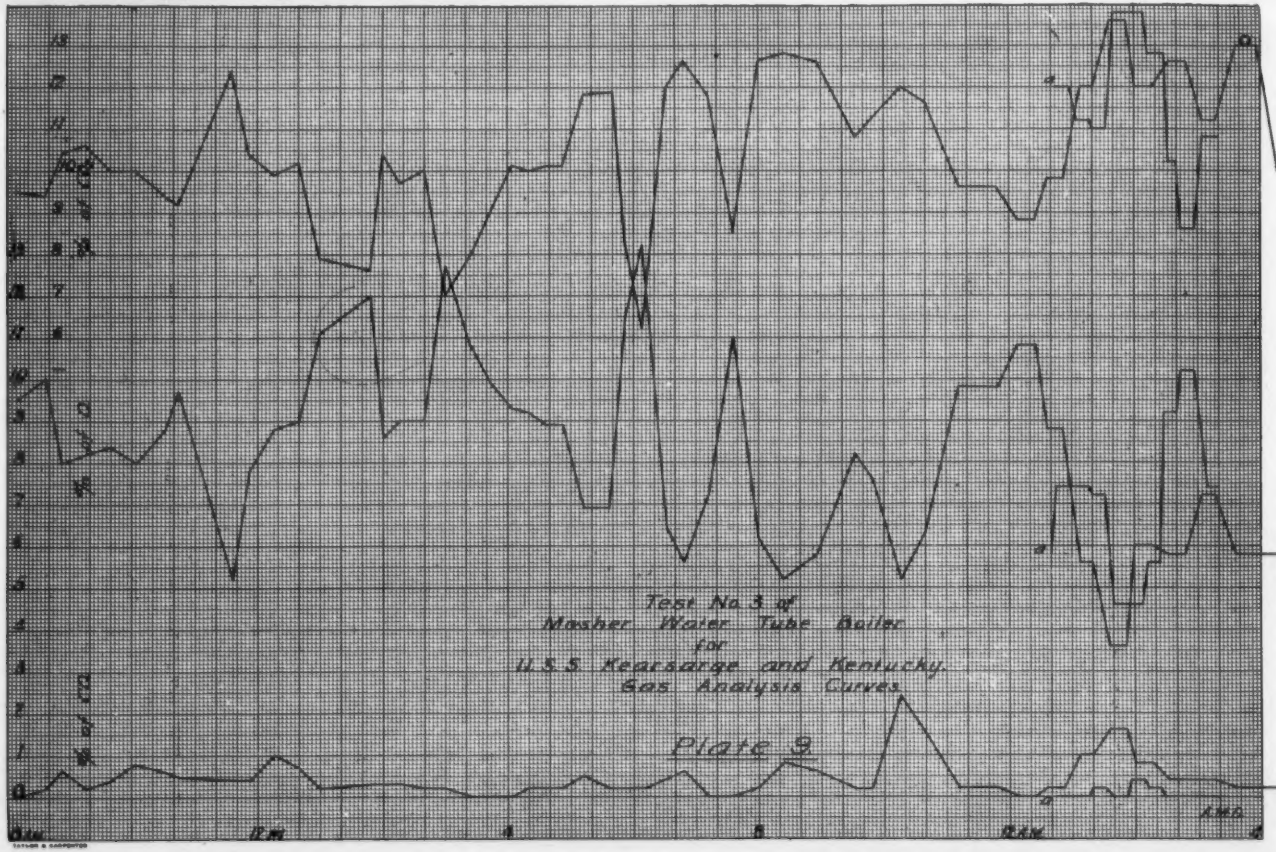


Plate 7
Test No 1 of
Mosher Water Tube Boiler
for
U.S.S. Kearsarge and Kentucky.
Gas Analysis Curves.


Test No. 2 of
Mosher Water Tube Boiler
for
U.S.S. Kearsarge and Kentucky
Gas Analysis Curves
Plate 8
a
b
c


Test No. 3 of
Mosher Water Tube Boiler
for
U.S.S. Kearsarge and Kentucky.
Gas Analysis Curves.
Plate 9.

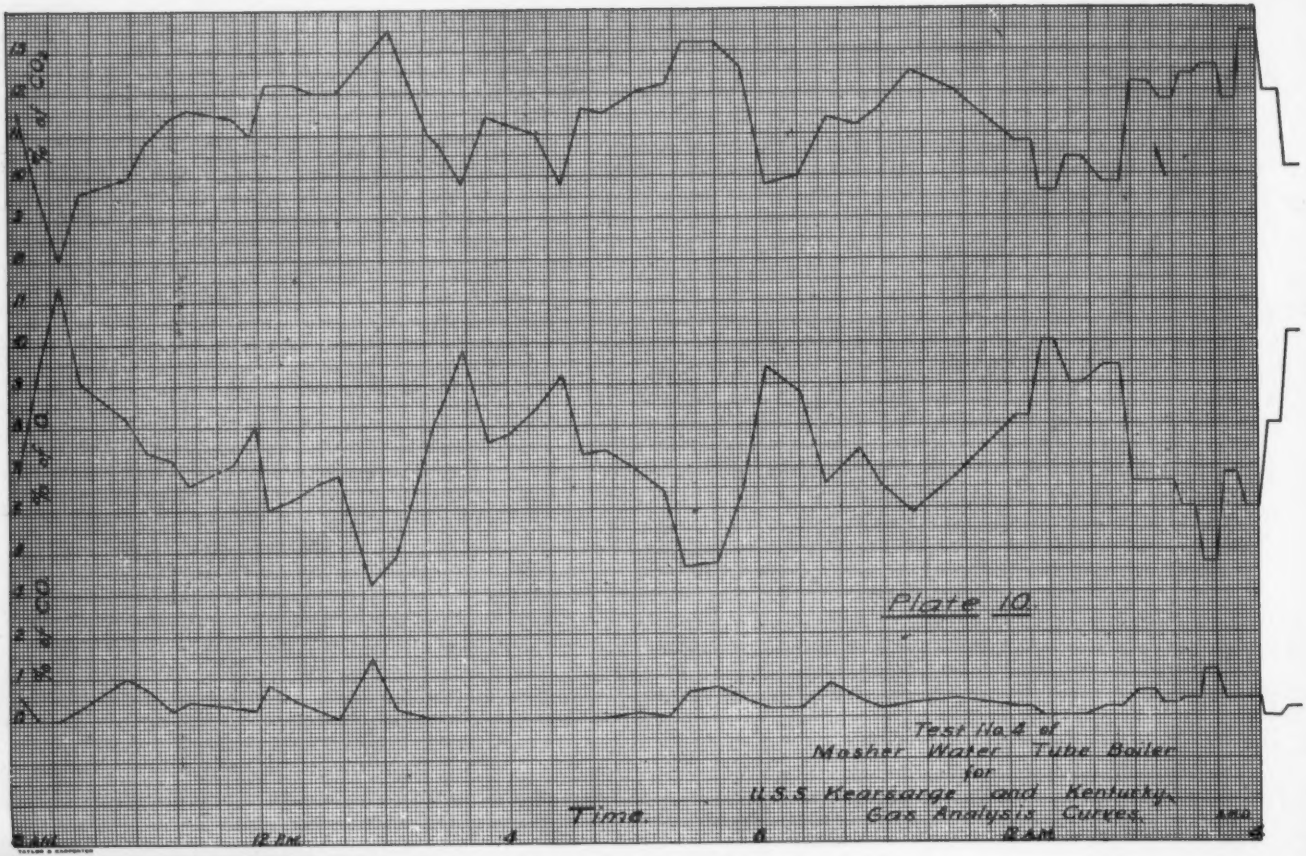


Plate 10.
Test No.4 of
Mosher Water Tube Boiler
for
U.S.S. Kearsarge and Kentucky.
Gas Analysis Curves.
Time.
12 PM.
4
8
12 AM.
4

NOTES ON DISTILLING FRESH WATER ON BOARD SHIPS.

By Lieutenant Commander E. H. DeLany, U. S. Navy, Member.

The question of a water supply on board modern vessels has recently become a subject involving the attention and study of the best engineering talent. The introduction of water-tube boilers on ships has necessitated an adequate supply as well as a good quality of fresh water.

The more extended cruising of naval vessels, culminating in the trip of the Fleet around the world, has shown in a marked degree the importance, if not the absolute necessity, that each ship should be able to maintain herself in water for drinking, washing, scrubbing paint work, make-up feed, and renewal of water after cleaning boilers.

There are few ports in the world where coal cannot be obtained, and there are but few ports where a supply of good fresh water can be had; and, with the exception of the supply of coal and provisions, there is no element that will enable a ship to remain away from its base for so considerable a length of time as her ability to make an adequate and cheap supply of fresh water.

The supply of water should be considered, first, as being adequate in quantity, and second, as cheap in production; the latter being required in order that the coal supply of the ship may be drawn upon as little as possible.

Subsequent to the invention of the Baird evaporator, and up to within the last ten years, very little, if any, improvement was made in the method of distilling. During the period of the general use of the Scotch boiler on naval vessels, distilled

water which was not considered pure enough to be suitable for drinking purposes was used in the boilers. The general introduction of water-tube boilers, however, has made it desirable that even for boiler purposes all water should be absolutely free from impurities.

The old-style evaporator installation usually produced water containing four, five or six grains of salt to the gallon, which water was considered perfectly good; but it has become apparent in the more recent developments in connection with the subject of evaporating that water with even less than a grain of salt can be produced by the use of separators in the vapor line, the degree of perfection in the distilling depending upon the efficiency of the particular type of separator used. And for the last four or five years it has been the constant endeavor of engineers to secure the most efficient separators possible, as it is realized that make-up feed water containing five grains of salt to the gallon will scale up a boiler just five times as quickly as the same amount of make-up feed water with only one grain to the gallon.

When it is considered that water with practically no salt and no oil will entirely prevent galvanic action in boilers, the importance of the use of distilled water as compared to shore water in ships' boilers will be somewhat realized. The provisions for guarding against the introduction of oil in boilers are very nearly perfected, and when the introduction of salt can be prevented to a large extent by the use of nothing but distilled water in boilers, scaling and deterioration of boiler tubes, due to galvanic action, will, to a large extent, be prevented.

The number of instances of ships requiring retubing, where the pitting of the tubes is largely due to the unavoidable introduction of salt into the boilers, is very great; and it is believed that the introduction of an adequate water supply on ships of the Navy, which will place them beyond the necessity of requiring water from shore, will, in all probability, be the means of doing away with the most dangerous factor to be considered in the deterioration of boiler tubes.

The general use of separators in the vapor lines has been a very marked feature in increasing the output of the evaporating plants in the service.

Evaporators equipped with efficient separators have been able to produce a normal output equal to twice the maximum rated capacity, at the same time producing a quality of water that averaged less than a grain of salt to the gallon. This was done on one battleship during a 30 days' test. The ship had a crew of about 600 men, and a "free spicket" was granted the crew, who were permitted to use as much water as they pleased and at any time they saw fit.

The evaporator in question was 3,750 gallons maximum capacity, and had a normal output of about 2,700 or 2,800 gallons. The vapor pipe was equipped with two separators, in series, the vapor passing from one separator into the other. The vapor pipe was 2½ inches in diameter, while the separators were about 15 inches in diameter. The vapor valves on the evaporator shell and on the distiller were carried wide open. The first separator was fitted with a trap on the drain leading to the bilge as well as with a water glass and a by-pass. The second separator was fitted with a water glass, and a drain, which was worked by the evaporator attendant by hand, in case any water appeared in the glass, which was seldom, as the first separator discharged the greater portion of the brine carried over by the vapor.

The priming of the evaporator, although never measured, amounted to about one gallon an hour, and was discharged from the drains of these separators. This was sufficient in amount to have contaminated all the water made during the test, had the separators failed to take it out of the vapor.

The steam to the evaporator was fitted with a reducing valve that had been altered so that it could be manipulated to change the pressure in the coils by a large wheel on the stem. The shell pressure was carried throughout these thirty days at about 30 pounds.

This test was made during a cruise in the West Indies, and on the trip home, much of the time being underway. It was

found that a shell pressure of thirty pounds, with the vapor valves wide open, could be carried under all conditions—rolling of ship—and in all localities. It was necessary when underway, and when the ship was rolling, to carry the water not higher than half way in the glass. When the ship was very steady at sea, or in port, it was perfectly safe to carry the water very near the top of the glass, so that all tubes were covered, and thus aided in evaporation. As near as could be ascertained, the covering of all tubes cheapened the cost of evaporation to a considerable extent.

The evaporator was blown down every four hours, and the time the evaporator was actually in operation was accurately noted.

The average output during the thirty days' trial was 375 gallons per hour for this evaporator, which was of 3,750 gallon size.

It was found that this one evaporator would keep a crew of over 600 men in drinking and washing water, as well as water for scrubbing paint work, when running on an average of 13 hours per day, while before the installation of separators in the vapor line it had required all possible output of two evaporators to keep the ship supplied with water. Moreover, it was noted as a peculiar fact that after the crew became used to the privilege of a "free spicket" the amount of water used was less than when the crew were on an allowance. This was accounted for by the fact that the crew when drawing water at specified times and on allowance, would take a bucketful, whereas when they were permitted to draw water at all times and in such quantities as they desired, each man took only such water as he actually needed. It was also noted that it helped the appearance of the ship to a marked degree, owing to the absence of buckets stowed away with water for future use.

Many devices have been introduced within the last few years to cheapen the cost of water. A few years ago two pounds of coal was considered to be the proper estimate of the cost of a gallon of water. With the introduction of water-tube boilers

and better firing devices the cost of producing steam was cheapened and the cost of a gallon of water was consequently reduced to about 1½ pounds of coal.

One of the most potent factors in bringing down the cost of water below this mark has been the introduction of the Bureau design of evaporator feed heater. This consists of a small feed heater with bent tubes, fitting into a shell, and having an outer casing for the vapor, the tubes being expanded into a tube sheet. The water fed to the evaporator is pumped around the tubes, and the vapor coming from the evaporator is passed through the tubes, the result being to utilize part of the heat of the vapor to heat the feed water going to the evaporator.

The revival of the double effect evaporating has also operated to decrease the cost to a considerable extent. The double effect and the evaporator feed heaters have been adopted so nearly together that it is impossible to make an accurate estimate as to how much credit is due each; but the combined result has been to reduce the cost per gallon of water to between three-quarters and a pound of coal.

Plate 1 shows a system of distilling that has been tried on the *Panther* and the *Yankton* with considerable success. The system consists in the installation of a high-pressure surface feed heater used as a distiller in the main feed line of the ship. It is a combination feed heater and distiller. The vapor surface is tinned and the heating agent is the vapor from the evaporator, no other vapor entering this feed heater. It is placed between the exhaust feed heaters and the boiler, and it is necessary that it be on the pressure side of the pump, the exhaust heaters being either on the suction or the pressure side. The economy results from the fact that the latent heat of the vapor coming from the evaporator, which ordinarily goes overboard in the cooling water passing from the distillers, is now returned to the boiler, being used to increase the temperature of the feed water going to the main boilers.

It was found on the *Panther* that a 5,000-gallon evaporator operated in this way could make about 8,000 gallons in port

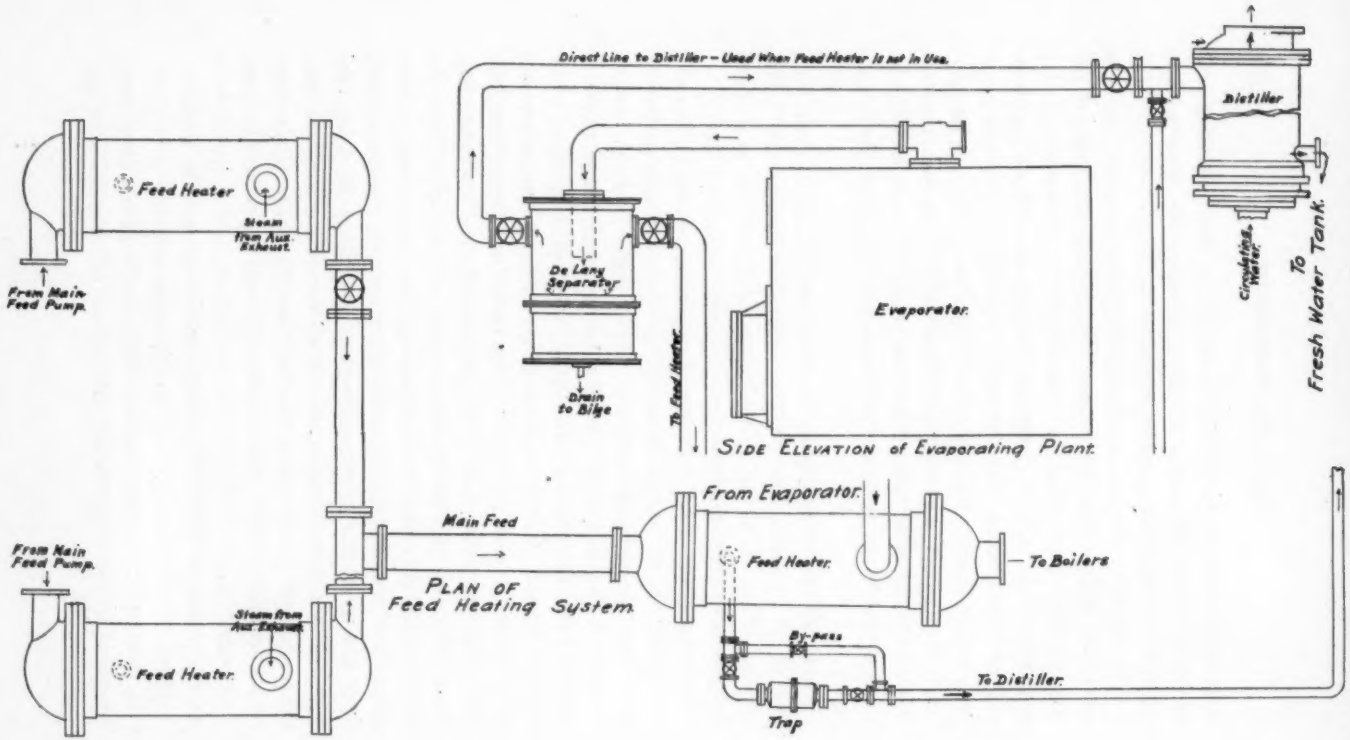


Plate 1.—DE LANY COMBINED DISTILLING AND FEED-HEATING SYSTEM.

and 12,000 at sea. The exhaust heating arrangement on the *Panther* is a jet in the hotwell, there being no surface exhaust heater, the feed water going into the distilling heater at sea being about 150 degrees, while the temperature coming out was about 200 to 225, depending upon the pressure carried in the shell of the evaporator. The evaporator was on the berth deck, and the vapor was led to this heater through a 3-inch pipe of sixty-eight feet in length; the drop in pressure in the pipe was about ten pounds, the drop in pressure in the length of the heater, which was eight feet, being about twenty pounds. The water was forced back to the distiller through an 1½-inch pipe, the force being the pressure in the shell of the heater. A by-pass in the engine room served to drain any excess water that accumulated in the heater to the feed tank. A trap was fitted in this line, but rarely operated. Any trap with a large discharge volume would be suitable, or a pump operated by a float would be sufficient.

It is necessary after the water has condensed in this heater to pass it through a distiller for the purpose of cooling it sufficiently for use. The salt water used around this distiller may, in the absence of an evaporator feed heater, be used to feed the evaporator. As nearly as could be ascertained the cost of water evaporated in this manner was about ¼ pound of coal per gallon, the expenditure of heat being that due to radiation in the system, also due to heat taken out of the condensed water by the distiller in cooling.

The amount of water condensed by a heater of a given size depends, of course, upon the amount of heat transmitted from the vapor to the water passing through the tubes; and this will depend upon the average difference in temperature between the vapor and the water; hence, the amount of water distilled may be increased by increasing the shell pressure on the evaporator, though this is limited by the amount of water passing through the tubes, and also by the temperature of the water as it enters the tubes. Increasing the amount of water passing through the tubes, or lowering the temperature of the same, tends to increase the amount of water that it is

possible for the heater to distil; but an increase in the amount of water going through the tubes, or a lowering of the temperature of the same, will not increase the amount of water distilled unless the coil pressure and shell pressure of the evaporator are increased to correspond with the increased cooling effect.

An attempt was made to install this system on two of the battleships, though in both of these cases the work was done while the ships were on the cruise around the world, and the time for efficient installation and trial could not be obtained. It was necessary to substitute the evaporator vapor for the exhaust steam in the heaters, so that the effect of the exhaust-steam heating was to a large extent thrown away, the only heating done being by the exhaust of the feed pumps into their own suctions.

It was found that in both instances the amount of condensation in these heaters was more than the ship needed in fresh water, and was also more than the means provided could take out of the heater, so that the heaters became water-logged.

It is believed that by installation of the heater as shown in Plate I efficient working of this system could be successfully produced on a battleship. On one of the battleships above referred to it was found that the system worked efficiently up to a speed of 10 knots, when the amount of water passing through the heater became so great that the evaporators were no longer able to supply vapor in sufficient quantity to produce a pressure in the heater. It is believed that with a little persistent experimenting this could be obviated by partially opening the by-pass around the heater so that the amount of water passing through the tubes would be just sufficient in quantity to condense the amount of water required for use in the ship. The distilling of any more water than that required would be, of course, a loss of ¼ pound of coal for each gallon of water so distilled, and for that reason is not a desirable condition.

On the *Yankton* and *Panther*, where this distiller feed heater was installed in addition to the jet heaters on those ships,

the results were very good under all conditions, although trouble occasionally appeared that had not been anticipated.

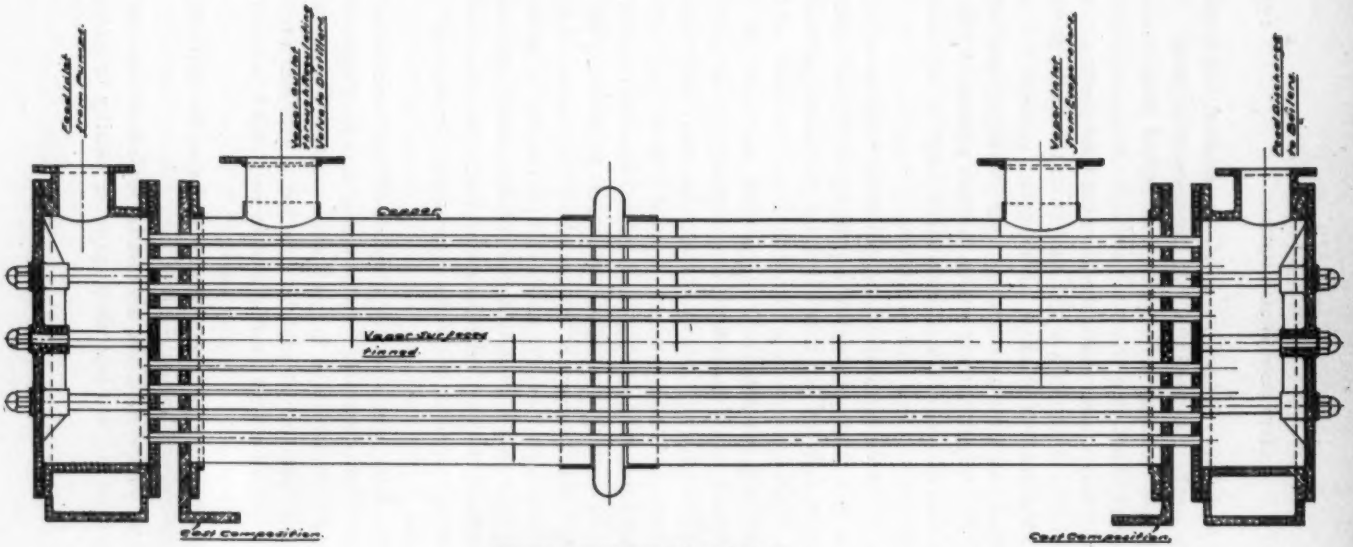
The heaters were made of copper, and copper tubes were used, but it was found that the tubes did not all expand alike, so that occasionally a tube would crawl in the tube sheet of the heater sufficiently to permit a slight leak from the water passing through the tubes (which is under boiler pressure) into the vapor space; and it was found that the castings when hot occasionally had slight defects which did not appear in testing, but which permitted the least bit of feed water to pass into the vapor space.

The leaks referred to would not ordinarily have been noticed or be of any consequence in the ordinary feed heater, but where it was sought to use the water for drinking purposes the least bit of oil or steam would contaminate this water and render it unfit for drinking. To avoid this the feed heater shown on Plate 2 has been designed, and though it has never been tried, it is believed it will entirely do away with this trouble. The heater as shown has a double tube sheet at each end, the tubes being expanded in each of the four tube sheets, and the lugs or feet, which secure the heater to its bed plate, have elongated holes to permit expansion of the heater. Any water that might leak from the water side of the heater, either through the tube sheet or around the tubes, would pass into the space between the two tube sheets, so that the only possibility of leaking water to the vapor side would be through a defective tube.

One of the prime requisites for good distilling is a constant feed, and for this purpose a duplex pump should always be used. The water should be carried at a constant level, and the vapor shell pressure should be constant. The coil pressure should always be controlled by a reducing valve fitted with a large wheel, and not by a stop valve.

It will be found that as the evaporator scales up the reducing valve must be opened out gradually.

Evaporators in constant use should be scaled at intervals from one to three weeks, depending upon the locality in which



FEED HEATING DISTILLER
FOR
DELANY FEED HEATING SYSTEM.
DESIGNED BY LIEUT-COM'DR. E.H. DELANY, U.S.N.

Plate 2.

the distilling is done. The scaling of an evaporator can be done by the evaporator crew, and if properly trained, this will become a matter of routine rather than a matter for the engineer officer's work book.

Sea water in the vicinity of the West Indies, or wherever coral formation is general and prevalent, is more difficult to distil from with safety than in other places.

The separators in use in the vapor line must not be considered as instruments for making the distilling apparatus foolproof, but only as an appliance that will enable careful attendance to produce good water under all conditions.

Thirty pounds on the shell is about the limit that can be carried under all conditions. In the mouths of rivers and bays, where the water is partially fresh, 40 pounds may be carried with safety.

Some engineers carry the bottom-blow valve partially cracked instead of blowing down at regular intervals. This is believed to be bad practice, for the reason that it is impossible to gage the loss of heat due to this constant blow. In evaporators which are blown down every four hours it has been found good practice to feed the evaporator through the bottom-blow opening; this prevents the possibility of scale gathering around this opening, the water constantly pushing the scale away; it also gives a more uniform rise of temperature in the water evaporated from the bottom to the top, which is essential in good, rapid distilling.

It is the experience of the writer that baffle plates in evaporators are of little or no use. It appears that the water which is deposited on these plates clings to them for a short space of time and is picked up by the vapor again.

The tubes of an evaporator should be beaded over to guard against the tendency of the tube to crawl out of the tube sheet when a very high coil pressure is used.

The installation of an efficient trap which will not permit the discharge of vapor is required to care for the water condensed in these coils. This trap should be preferably fitted with a gage glass so that the attendant may be sure that no

water remains in the coils or in event of this occurring the trap may be cleared by means of the by-pass.

One hundred and twenty-five pounds of coil pressure is about the limit that should be required in the coils before the evaporator needs scaling. The use of this high-pressure steam is but very little more expensive than the use of low-pressure which is required when the coils are clean; hence, the evaporators should be used without scaling until it appears that the tubes are so thickly coated as to reduce the output of the evaporator, which will be when about 125 pounds are necessary to produce 30 pounds on the shell.

In the vicinity of the West Indies an evaporator used constantly will get into this condition in a week or ten days.

The coil pressure necessary to run when the tubes are clean is about 40 pounds.

By observing the output of different installations the writer has come to the conclusion that, while separators have increased the assurance of making fresh water up to and a little above the maximum rating of evaporators, no remarkable increase in the output of an evaporator has occurred where two or more evaporators have been operated in parallel. The reason for this will be readily understood when it is considered that evaporators are like a battery of boilers connected in parallel, but on account of being fed with salt water are much more sensitive and liable to foam; and the instant that one evaporator commences to make vapor faster than the others it will siphon water from them to a certain extent, and cause priming to a degree that separators are not able to take care of.

It should be possible to connect an evaporator to at least one other distiller, to be used in case the regular distiller is out of commission.

It is beyond doubt that the greatest output from an evaporating plant is to be obtained when each evaporator is connected by a vapor pipe to its own distiller and ordinarily has no connection whatever with the other plants. Piping in this way has been abandoned to a certain extent with the idea of cheapness but it will eventually be resumed, because the ad-

ditional cost of the piping and valves required to enable an evaporating plant to operate so that each evaporator connects to its own distiller is very slight. A simple system of piping for this arrangement is to have one additional distiller. The vapor pipe from each evaporator should be led up between two distillers with a valve leading to each. This would permit each evaporator to work with either one of the two distillers. It would also permit each evaporator to work on a separate distiller when all are in working order.

The increase in efficiency in many plants having more than two evaporators, where the second-effect arrangement has been installed, is to a great extent due to the fact that the use of evaporators in series has done away with their use in parallel; but it would be found that these same evaporators when operated singly and as independent plants, will more than triple their normal output, provided they are equipped with efficient separators.

The use of double-effect evaporators, as near as the writer can learn, will produce almost the same amount of water as the two evaporators would produce if run in parallel and kept down to an output which will prevent their siphoning each other.

The high-pressure evaporator will produce considerably more than it will if run in parallel; while the low-pressure evaporator will, of course, produce somewhat less. But the output of the high-pressure is limited by the power of the low for condensing it. For this reason where ships have three evaporators, if the double effect system is introduced it will be found advisable to run with one evaporator as the high and the other two as low pressures. This will produce the maximum economy in the system and enable the high-pressure evaporator to largely increase its output, due to the fact that ample condensing surface is thus provided. A separator on the high-pressure vapor line or on the vapor line leading to a feed-heating distiller is increased in efficiency for the reason that its drain discharges more easily. This is due to the increased pressure in the vapor line.

The adoption of the system shown in Plate I will necessitate the evaporators and distillers being placed between the engine and firerooms in the space now occupied by coal bunkers, and the use of the space now used for the evaporators for a reserve coal bunker.

Of course this system could be adopted by passing the vapor pipe through the protective deck to this distilling heater, but this is not advisable, because it requires the care and service of men in the evaporator room and another man in the engine room to make its working efficient, and this is hard to attain unless the intention and determination to make the system work is very marked, which is hardly ever the case with the introduction of anything of a new nature aboard ship. But with the placing of the evaporators and distillers as well as the feed heater altogether, the matter becomes very simple.

The placing of the distillers in the engine-room hatch or other location of considerable height above the evaporators, is no longer necessary since the introduction of separators; and from a military standpoint is very undesirable, as there would be no time when water would be more urgently needed than immediately after a battle or a long run; moreover, it is not desirable that the plant should be fitted in an exposed position.

The distillers should be fitted with air cocks, and any evaporating plant having separators in the vapor line should have these separators fitted with snifting valves to prevent any possible vacuum being formed in the separators. The use of a vapor line connecting to the auxiliary exhaust line or to any of the condensers should be entirely abandoned, for the reason that it is impossible to tell whether the water so being distilled is fresh or not until the damage has been done; and, secondly, because it is impossible to use a separator on a line having a partial vacuum, for the reason that the separator will not discharge unless the pressure within it is greater than the atmosphere.

In installing separators care should always be taken to see that they are properly located. They should be installed a lit-

tle higher than the top of the evaporator and as near to it as possible. A considerable length of piping should connect the separator to the distiller in order to prevent the condensation in the distiller from forming a partial vacuum in the separator. The position of the distiller as to height is not important.

When the output of the evaporating plant on a ship in the Navy has increased to any marked degree, it will appear at first that the distillers are not sufficient in capacity to make cold water. This may be corrected by setting aside one distiller, closing the valve connecting it to the vapor pipe, and then connecting it to the discharge of the distributing pump, so that the water being pumped to the tanks is passed through the vapor space of the distiller, the piping being arranged in such a way that the vapor space will always be full of fresh water en route to the tanks. After blowing down it will be found that the evaporator makes bad water for about five or ten minutes after starting up. For this reason every plant should be fitted so that the distilled water may be discharged into the bilge, and this valve should always be open before starting up the evaporator. The water should be discharged into the bilge until the tests show that it is good.

There should also be a cock on the discharge from the distillers for continuous testing, and the medical department of every ship should be equipped with necessary apparatus for quantitative analysis of water.

NOTES ON LEAD STORAGE BATTERIES FOR SUB-MARINE USE.

By J. B. Howell, Ensign, U. S. Navy, Member.

This article is largely a compilation from the works of E. J. Wade, Treadwell, and Lyndon, and is intended to place before the officers and enlisted men of submarine vessels the theory and practice of Storage-Battery Engineering.

Principle of Storage Battery.—The underlying principle of a storage battery or accumulator is the property that certain elements have of possessing a potential difference when placed in an electrolyte. Two such elements placed in a hydrogen containing liquid will undergo an electro-chemical change, *i. e.*, one will give up certain of its properties to the other.

Elements Used.—The most successful elements used for this purpose are pure sponge lead and lead peroxide. These two metals have a potential difference of 2.0 volts on open circuit and 2.7 volts on charging circuit.

Method of Using Elements.—The lead-peroxide plate is used as the positive and the pure sponge lead as the negative. These elements are immersed in an electrolyte of sulphuric acid, H_2SO_4, of 1.20 to 1.40 specific density, and held apart by separators of hard rubber or wood. Upon the connection of these plates to an outside circuit an electric current will flow, and the following internal chemical changes take place:

Positive plate, $PbO_2 + H_2SO_4 = PbSO_4 + H_2O + O$.

Negative plate, $Pb + O + H_2SO_4 = PbSO_4 + H_2O$. Or in words, both electrodes are changed into lead sulphate, with the extraction of sulphion SO_4 from the electrolyte and the formation of water.

Chemical Reactions.—The reaction is considered thus: first, the reduction of the peroxide of the positive to monoxide and

the conversion of the monoxide into lead sulphate. In this process two atoms of oxygen are liberated, one is taken up to form lead sulphate and the other goes over to the negative and forms monoxide with the pure sponge lead; this in turn extracts SO_3 from the electrolyte and forms lead sulphate on the negative; all of these reactions are shown as follows:

Stage.	Positive.	Electrolyte.	Negative.
1st	PbO_2	$H_2SO_4 + H_2O$	Pb
2d	PbO	$H_2SO_4 + H_2O$	PbO
3d	$PbSO_4$	$H_2SO_4 + H_2O$	$PbSO_4$

If this process is carried on at too rapid a rate the electrolyte is decomposed in such manner that a new compound is formed which corrodes the positive plate.

The individual battery plates are assembled together to form a complete cell, by "burning" to a common lead bar or strip, all the plates of like polarity; there being, of course, two bars for each cell, one joined to the positive and the other to the negative plates. The connection for leads to switchboard is a matter of design and differs with type of cell. Provision should always be made to keep the battery well ventilated.

Assembling of Cells.—The cells are then joined together. Perfect insulation of cell jars and entire battery is required to give good results.

Great care and excellence of workmanship is required to produce a battery without having large voltage leakage. In assembling plates, joining the edges of lead parts or making any lead joints, the two pieces to be joined should always be flowed or "burned" together.

Installation.—The burning consists of subjecting the edges to be united to a high temperature, such that the lead is melted and flows together, forming in effect a lead weld. This is accomplished by means of the oxy-hydrogen blow pipe. All bolt connections or other metal parts should be coated with lead.

The envelopes or separators which have been found successful are: hard rubber, celluloid, asbestos cloth, glass wool and

pyroxylin. Hard rubber in either case or sheet form, strengthened at intervals by ribs, with a large number of perforations, is best. Some companies use wood separators. They seem to be effective and efficient, but there is always danger that acetic acid may be released from these and injure the plates.

Jars.—The best containing jars are made of hard rubber or lead-lined wooden receptacles; the plates rest on glass supports at ends of jars.

Character of Plates.—The character of the active material best suited to produce long life is hard and tough, yet porous and of low specific gravity. The positive plate should resist the pressure of the thumb nail while the negative should be soft and easily pressed into by the nail.

Electrolyte.—Density of electrolyte should not be too high or there will be danger of sulphation; but there should be plenty of it in the cell to completely cover the active parts of the plates. Highly dilute solutions will, however, have a deleterious corrosive effect which will shorten the life of the elements. Impurities in the electrolyte corrode and eventually disintegrate the plates.

Charge, Rate of.—The rate of charge should be moderate to keep the electro-chemical changes within moderate limits.

Charging.—The practice is to use the eight-hour rate, *i. e.*, eight ampères for every square foot of plate area. At this rate 50 per cent. of the charge is absorbed at the end of the first hour, 75 per cent. at the end of the second and 85 per cent. at the end of the third; the remainder is taken up at a slow rate. The color of the plates indicates their condition of charge, the positive should be of a dark slate and the negative of a light slate at end of charge.

The voltage at the normal charging rate should show 2.0 on the beginning, rise rapidly to 2.3, where the evolution of hydrogen gas begins at the negative, and then rise slowly to 2.6, where gasing of both plates occurs.

Local Action, Partial State of Charge.—It must be impressed upon the operators of storage batteries that the local action going on in a battery is greatest when in a state of par-

tial charge. This action will be indicated by the negative plates lagging behind on charge. It is obvious that the battery should be kept in a charged condition, and if any cells show local action they must be attended to.

Overcharge.—It is occasionally advisable to give the entire battery an overcharge, but not often, as the gases formed inside the pores tend to crack or peel off the active material.

Temperature on Charge.—The maximum temperature of battery on charge should not exceed 100 degrees F.

Capacity Limit.—As regards capacity, the battery should never be discharged beyond 75 per cent. of the amount of charge put in.

Discharge, Rate of.—Moderate rate of discharge is desirable owing to the fact that material is rapidly contracting, and the negative plate has a tendency to give off some of the pure lead as lead oxide which would produce local action.

Discharge Voltage.—The minimum voltage per cell on discharge is the most important of all factors; it is 1.85 on slow rate and 1.75 on rapid rate.

Capacity, Loss of, and Remedies for.—*Loss* of *capacity* may arise from clogging of pores, contraction of pores, loss of active material, formation of a layer of sulphate between the grid and the active material, or loss of electrolyte. When the negative plates show such a loss they should be placed in a bath of 1.200-density sulphuric acid, and connected as positives, the negative being a "dummy" plate of plain sheet lead about one-sixteenth inch thick. This cell is then charged; the sponge lead of negative first turns into PbO then PbO_2. By reversing the current the lead becomes completely peroxidized and we have pure Pb on the negative. It is sometimes possible to treat the whole battery in this manner by reversing the charge at about one-half the eight-hour rate; this, however, is not good practice.

The plates must be efficiently separated to prevent internal discharge, as this causes over-sulphation.

Sulphation.—The causes of *abnormal sulphation* are: over-discharge or discharge at rapid rate; intentional through ex-

ternal circuit; local action leakage; loosening of active material, or short circuit between plates. Over-sulphation may be detected by the gradual lightening of the plates in color. The best method for treatment of these plates is charging at one-quarter normal rate for a long time, or at a rate which will not run temperature above 105 degrees F. When sulphate once forms several cycles of charge and discharge are necessary to bring up battery to normal capacity.

Corrosion of Plates and Remedies.—*Corrosion of plates* may occur from two causes: the chemical action of highly dilute acid in the pores of plate, or impurities in the electrolyte.

The first condition cannot be remedied, as it occurs if the discharge is carried too far or if the plates have a thick layer of active material when discharge rate is high. If the electrolyte contains lead dissolving salts their presence will be detected by increase of capacity, which means the attacking of the grid. The obvious remedy is to change the electrolyte.

Fracture of Plates, Remedy.—*Fracture of plates* indicates too rapid discharge or non-uniformity of the plate material. If the trouble arises from defective plate there is no remedy; if from discharging at too rapid a rate the remedy is obvious.

Shedding of active material cannot be prevented if plate is improperly made; it can be lessened, however, by charging at a lower rate and not going above 2.4 volts per element.

Outside Plates.—The outside positive plates of a cell disintegrate before the others, because they are more active and give off more than their proportional amount of current.

Reversal of Negatives.—Reversal of negative plates is due to complete discharge below zero, or overpowering of defective cells. The remedy is, of course, to recharge the low cells and continue the charge until cell is up to normal capacity.

Loss of voltage is of frequent occurrence; it manifests itself by various cells being lower than others in battery; it is due to over-sulphation. These cells must be treated for same.

A good battery will show all cells of the same voltage and the same density of electrolyte. When charging, if all cells

are gasing at the same time the operator can feel safe in the knowledge that his battery is in a good condition.

Electrolyte, Loss of.—Loss of electrolyte from evaporation should be made up by the addition of about 95 per cent. pure distilled water and 5 per cent. acid.

TESTS.

Discharge.—When the discharge is made at variable rates and does not extend over two weeks, it should be stopped when the specific gravity of the pilot cells is from 40 to 50 points below that obtained on previous overcharge; and if service conditions are such that battery is not discharged to this limit the battery should be given a discharge to that state.

Charge.—The charge should begin immediately after the discharge is completed. It should be begun at the normal rate until the voltage begins to rise rapidly, when the voltage should be diminished to the half rate until the end.

Temperature.—The temperature of pilot cells should never be allowed to go above 110 degrees F. This is in accordance with the theoretical limit, and any increase in temperature will only injure the battery.

Ventilation.—The ventilation should be regulated so that gases cannot collect over the cells.

Electrolyte.—The electrolyte should be kept from one-quarter to three-quarters of an inch above the top edges of the plates.

Readings, Records and Reports.—The following readings should be taken on each charge and recorded: charging rate, battery voltage, and specific gravity and temperature of pilot cells. These readings should be taken at the beginning of charge and every half hour at normal rate, every 15 minutes on the half rate.

Daily readings of voltage of both batteries, ground voltage of both, and the specific gravity of electrolyte of pilot cells.

Yearly Overhauling.—The battery should be given a dis-

charge test just before the annual overhauling. Cadmium tests should be taken in addition to the other routine readings.

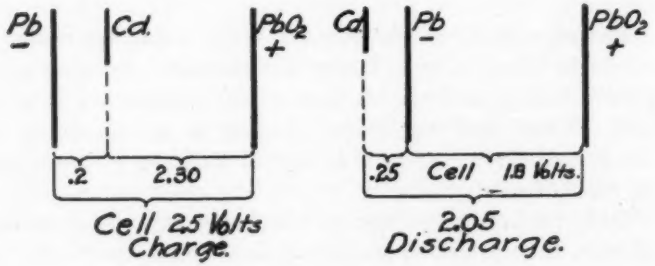
Cadmium Tests.—The maintenance of the voltage, and consequently the capacity of the cell, depends on both the positive and negative plates, and, therefore, if one be fully charged or fully reduced and the other is only imperfectly charged the capacity is small.

The voltage of a battery is not always an indication of the state of charge, and in order to determine the condition of the two plates it is necessary to test them independently. This is done by immersing cadmium in the electrolyte and observing the voltage between it and the positive and negative elements.

The cadmium must be free from marked impurities, and surface should never be scraped bright.

When a battery is discharged down to 1.8 volts the voltage between cadmium and the positive plate should be about 2.05, and between cadmium and the negative plate .25, cadmium being positive to both elements in the voltaic sense. Voltage of cell is 2.05—.25 = 1.8, the cadmium negative reading being substracted from the cadmium positive reading.

When battery is charged, cadmium to positive plate about 2.35 volts, cadmium to negative plate .18 to .20 volts; cadmium being positive to positive plate and negative to negative plate; that is to say, voltaically considered. Voltage of cell is 2.35—(— .20) = 2.55.



Condition of Battery.—The battery as a whole must be kept in an immaculate condition. All parts of the exposed headers and cell jars should be washed with KOH from time to time; all bolt connections kept well coated with insulating paint, and all lead burning examined.

Don'ts.—There are a few handy don'ts given below:

Don't add acid to electrolyte to make up for evaporation;

Don't discharge battery below 1.80 volts per cell;

Don't charge above normal rate;

Don't charge at normal rate after highest element is 2.6 volts, but tail the charge off.

DEVELOPMENTS IN WIRELESS TELEGRAPHY.

By Commander S. S. Robison, Member.

Park Benjamin, in his history of "The Intellectual Rise in Electricity," traces the idea of the telegraph as far back as the fifteenth century. He wrote in 1895, at or about the time that Marconi was making his first experiments in wireless telegraphy, and the historical instances he adduces of the existence in men's minds of the idea of communicating at a distance apply more directly to wireless telegraphy than to the Morse telegraph, to which he referred.

The experimenters and philosophers of those days hoped to achieve this communication by means of sympathetic magnets instead of tuned or sympathetic wires.

We read that an Italian, John Baptista Porta, in a book on natural magic, published in 1589, states "That to a friend that is at a far distance from us and fast shut up in prison we may relate our minds" by means of "two mariner's compasses having the alphabet writ about them."

This idea seems to have persisted, and in 1600 a German, Daniel Schwenter, undertook to make a working apparatus on this principle. Attention to the working of the two compasses was to be called by the receiving compass needle striking a small bell placed in its path. It was a necessary feature of these sympathetic needles that both should have been stroked by the same lode-stone—we now say tuned to the same wave length.

They even had promoters in those days who were quite as eager to get something for nothing as those we have to contend with. Benjamin states that Galileo dealt with them in a way which has served ever since as an example to be followed by the sceptical capitalist besieged by the sanguine inventor,

"You remind me," Galileo makes one of his characters say, "of a man who wanted to sell me a secret of communication by the sympathy of magnetized needles so that it would be possible to converse over a distance of 3,000 miles. I told him I would willingly purchase it provided he would show me an experiment, and that it would suffice if I remained in one room while he went into another. He replied that the distance was too short to exhibit the operation of the invention properly, so I dismissed him, saying that it was not convenient for me to travel just then to Cairo or Moscow to test the matter, but that if he would go there himself I would remain in Venice and do the rest."

In 1646 this thing "was whispered through the world with some attention, credulous and vulgar auditors readily believing it and more judicious and distinctive heads not discarding it." The idea served as a charming conceit for literary purposes in the next century.

The foundation for real wireless telegraphy was laid by the mathematical deductions of Sir William Thompson relative to the oscillatory discharge of a Leyden jar; the theory of Maxwell that light and electricity were manifestations of the same nature differing only in their period of vibration; and the experiments and measurements of Hertz, by whom the theories of Maxwell were proved to be true.

The *idea* was expressed anew with more completeness by Sir William Crookes in 1892, three years before Marconi made a working reality of what until that time was only an idea.

Crookes wrote as follows: "Whether vibrations of the ether longer than those which affect us as light may not be constantly at work around us we have until lately never seriously inquired, but the researches of Lodge in England and Hertz in Germany give us an almost infinite range from wave lengths of thousands of miles down to those of a few feet. Here is unfolded to us a new and astounding world, one which it is hard to conceive, should contain no possibilities of transmitting and receiving intelligence.

"Rays of light will not pierce through a wall, but electrical

vibrations of a yard or more in length will easily pierce such media, which to them will be transparent. Here is revealed the bewildering possibility of telegraphy without wires, or any other of our present appliances. Granted a few reasonable postulates the whole thing comes well within the realms of possible fulfillment.

"At present experimentalists are able to generate electric waves of any desired length and to keep up a succession of such waves radiating into space in all directions.

" It is possible, too, with some of these rays, if not with all, to refract them through suitable bodies acting as lenses and so to direct a sheaf of rays in any given direction. Also, an experimentalist at a distance can receive some, if not all of these rays, on properly constituted instruments; and by concerted signals, messages in the Morse code can then pass from one operator to another.

" What, therefore, remains to be discovered is:

" Firstly, *Simpler and more certain means of generating electrical waves of any desired wave length.*

" Secondly, *More delicate receivers, which will respond to wave lengths between certain definite limits and be silent to all others.*

" Thirdly, *Means of darting the sheaf of rays in any desired direction,* by the help of which the sensitiveness of the receivers (apparently the most difficult of the problems to be solved) would not need to be so delicate as when the rays to be picked up are simply radiating into space in all directions and fading away according to the law of inverse squares."

This clear exposition of the needs of wireless telegraphy, clear today, probably attracted momentary attention as a scientist's dream and was then forgotten.

Crookes even went farther and pointed out some of the methods that should be employed and the difficulties that might be in the way. For instance, he states that if the exact position of both sending and receiving instruments were known the third condition (directed waves) could be profitably employed, but if the sender and receiver were moving about that this

would not serve the purpose, since neither one would know the direction of the other, and that then the method to be employed is that the correspondents must attune their instruments to a definite wave length, say 50 yards, and that thus attuned the receiver might respond to waves varying between 45 and 55 yards and be silent to all others.

He assumed that the progress of discovery would give instruments capable of adjustment by turning a screw or altering the length of a wire so as to respond to waves of any preconcerted length and that, considering the wide range of wave lengths to choose from, there would be no difficulty in ensuring secrecy and freedom from interference.

The purpose of this article is to show as briefly as may be to what extent wireless telegraphy has developed in the nineteen years which have elapsed since Sir William Crookes' prophecy was uttered, using as a basis the three requisites which he sets forth as remaining yet to be discovered.

Firstly, *simpler and more certain means of generating electric rays of any desired wave length.*

Marconi in his first experiments used what is known as a Righi oscillator, generating waves of approximately 12 centimeters in length, with which he successfully transmitted messages up to a distance of two miles in 1895, the oscillator being placed in the focus of a parabolic mirror.

He presented as the concrete results of these experiments a transmitter consisting of two brass spheres 11 centimeters in diameter, placed one milimeter apart, which when oscillating generated about 250,000,000 waves per second, each about 1.3 meters long. This apparatus is diagrammatically illustrated in figure 1.

In the summer of 1896 with an apparatus similar to this he demonstrated the operation of his system over a distance of four miles for the British postal authorities, and on June 2, 1896, he made application for a patent. He completed the specification accompanying this application in March, 1897, and the patent was granted on July 2, 1897, about one year after the application was made.

The provisional specification, under date of June 2, 1896, indicates a generating apparatus like that in figure 1.

The completed specification presented in March, 1897, shows a transmitting apparatus, illustrated in figure 2, which indicates a means of generating electric waves of a much greater length than that shown in figure 1.

The arrangement shown in figure 2 originally included oscillating brass spheres between the two spark-gap points, but Marconi states in his specification that these are unnecessary, and this is a fact.

The arrangement shown in figure 2, he stated, would enable one to signal through obstacles such as hills or mountains, or across them.

He had measured and knew the lengths of waves produced by the oscillating spheres, but as far as indicated by the patent specifications he did not know the lengths of the waves produced by his modified apparatus for signalling, as he states, through hills or mountains. Although no attempt was apparently made to actually ascertain the lengths of the waves generated in his subsequent experiments, still it is shown that he was familiar with the fact that the wave lengths generated by a vertical earthed wire were approximately equal to four times the length of the wire.

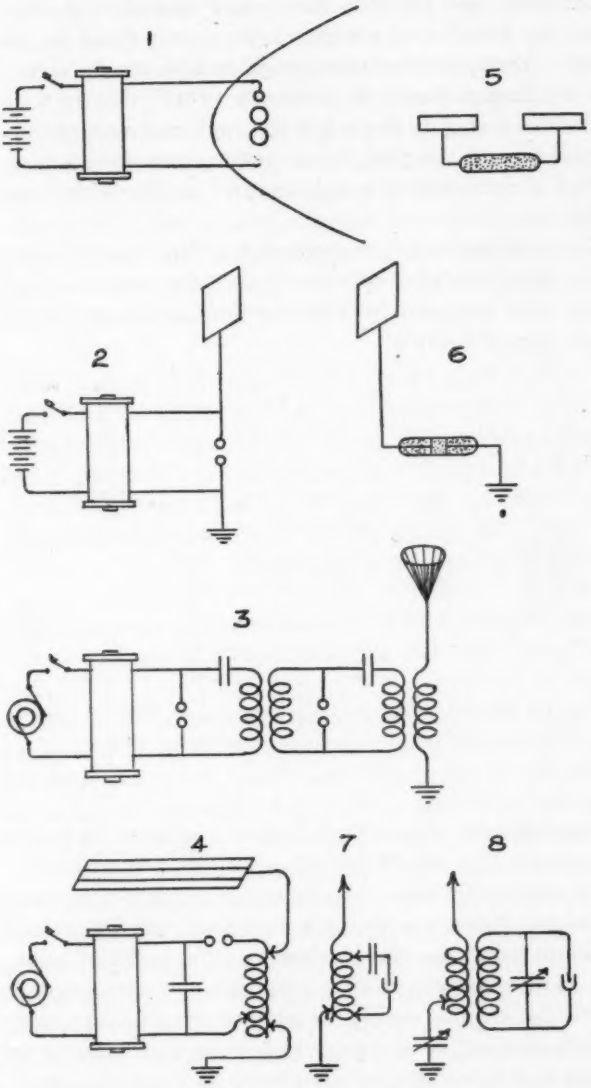
Before his patent was granted he made trials for the British Government in 1897, using the apparatus shown in figure 2, the vertical wires being 150 feet high in one case and held up by kites at a height not given in another case.

Signals were successfully exchanged at a distance of about 9 miles.

During the next year a distance of 34 miles was reached, and in 1899 communication was successfully established between England and France, about 32 miles, and between two British men-of-war about 60 miles apart.

In another case communication was carried on between shore stations with masts 150 feet high and 85 miles apart, the distance being partly over sea and partly over land.

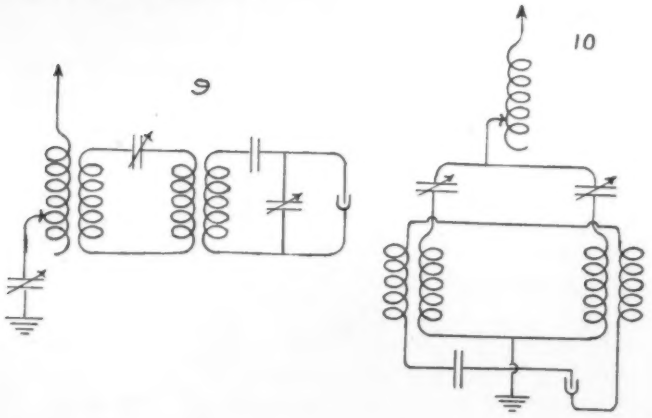
In 1899 Marconi came to America to report the international


1
5
2
6
3
4
7
8

yacht races held off New York, and subsequently his apparatus was installed on several vessels of our Navy for experiment. Communication was carried on between the *New York* and the *Massachusetts* at a distance of 47 miles.

In 1900 a ring of masts 200 feet high was built by the company formed to exploit his invention, and these masts supported a cone-shaped aerial, shown in the upper part of figure 3.

The arrangement for producing electric waves from this aerial were decidedly different from those shown in figure 2. They were designed by Fleming and are illustrated in the lower part of figure 3.



An alternator instead of a battery was used for producing the current supplied to the induction coil, and instead of the spark gap being placed directly in the aerial it was connected in another circuit containing a condenser, and this circuit, by means of what was then known as a Tesla transformer, but now more generally called an oscillation transformer, transferred the electric energy to another circuit also containing a spark gap and condenser, which in its turn transferred the energy to the cone-shaped aerial and set it into vibration.

It required about one year to construct and install this ap-

paratus, which used 25 horsepower, the frequency of alternating current being 50 cycles per second.

The station in which this apparatus was installed was built in the vicinity of Poldhu, Cornwall, England, and with it and from it signals were sent across the Atlantic and received on an aerial about 400 feet long, suspended from a kite near St. Johns, Newfoundland, in December, 1901.

In December, 1902, a similar station was completed at Cape Cod, Mass., and on January 19, 1903, a message from the President of the United States to the King of England was sent to Poldhu, which read:

"To His Majesty, Edward VII, King of England.

"In taking advantage of the wonderful triumph of scientific research and ingenuity which has been achieved in perfecting a system of wireless telegraphy I extend on behalf of the American people most cordial greetings and good wishes to you and to all the people of the British Empire.

Signed, "THEODORE ROOSEVELT."

While the wave length used for this tran-Atlantic communication is not given it was probably not less than 1,500 meters.

So that in less than eight years after wireless telegraphy was shown to be practical a range of wave lengths from 1.3 up to 1,500 meters had been used. The means of generating waves had been radically changed, but they were not made more simple, as Crookes said was desirable, though both the kind and amount of electric power utilized and the method of transforming this power into electric waves had been changed.

There is no evidence to show that any particular wave length was determined upon for any of the stations built at this time, nor any definite means of determining what wave lengths were being generated.

The method of construction shown in figure 3 was during the course of the experiments modified so as to omit one of the oscillation transformers, and with this omission this

method of generating electric waves is still in use and is commonly called that of inductive coupling. It was introduced in Germany about the same time by Braun.

Prof. Slaby, who had assisted at the Marconi experiments in 1907, subsequently produced in collaboration with Count Arco what was known as the Slaby-Arco system, which used the method of direct connection or direct coupling shown in figure 4. This became the general method of coupling transmitters and was used by Marconi, Fessenden, DeForest and Shoemaker, and other American inventors except Stone, who used the method of inductive coupling both for sending and receiving.

One of these methods, either direct or inductive coupling, as distinct from the plain aerial illustrated in figure 2, is universally used in wireless telegraph transmitters.

The inception of the direct or inductive method of connection as distinguished from the plain aerial was apparently due to the fact that a Leyden jar placed across the spark gap of a plain aerial added to the strength of current that could be produced, to the volume of the spark, and to the strength of the signals and consequently the distance at which they could be received.

In measuring the oscillating currents produced in the aerial it was found that different adjustments of the circuit consisting of the condenser and spark gap gave different readings of the hot-wire ammeter, other features remaining the same. When means of measuring the lengths of the waves generated were produced it was found that the readings of the hot-wire ammeter reached a maximum when the oscillations created in the circuit composed of the condenser, its connections and the spark gap (which form the closed circuit), had the same period as those produced in the aerial when earthed, and which is called the open circuit.

This indicated the desirability of tuning the open and closed circuits to each other when direct or inductively-connected transmitters were used.

The first portable and convenient means of measuring wave

lengths produced by oscillating circuits were made by the Slaby-Arco Co. in Germany about 1902, and were known as Slaby measuring rods.

Scheller and Count Arco also produced a wave meter made from the receiving coil of the Slaby-Arco apparatus.

The use of these wave meters showed that, in general, circuits tuned to resonance and connected to each other as shown in figures 3 and 4, generated and radiated waves of two lengths, one longer and one shorter than that which would have been produced by either circuit if oscillating free from the influence of the other. But that if the direct-connected circuits had a very small amount of wire in common, or the inductively-connected circuits had their coils removed to a considerable distance from each other so that the energy was transferred between them comparatively slowly, then these two waves almost merged into one.

Thus, a difficulty confronted the designers of transmitting apparatus. If they used the plain aerial and generated waves of but one length only they were limited to small powers and consequently short distances.

If they used direct or inductively-connected tuned circuits, they generally produced waves of two lengths, only one of which could be received, and part of the energy was consequently wasted.

The well known formula for the work done in charging a condenser: $\frac{CE^2}{2}$ shows that in the ordinary aerial, which has a capacity of less than one standard Leyden jar, comparatively little energy can be stored except at a voltage so high as to be inefficient. Therefore, unless the frequency of the charge and discharge (alternator frequency) is very high only small amounts of energy can be radiated from a plain aerial, whereas the condenser in a closed circuit can be made as large as desired, and therefore capable of absorbing and transferring to the open circuit any amount of energy. It not being necessary to store in the aerial all the energy for any one alternation of the sending current before discharge takes place, but

only to transfer a certain amount of this energy from the closed circuit to the open circuit of the aerial during each oscillation, this energy being radiated in the shape of electric waves.

So that, although the means were discovered of generating waves of any desired length and radiating any desired amount of power, difficulties arose in the use of the large powers which were found to be necessary to send long distances—two waves were generated where only one was desired. This difficulty was partially overcome by weakening the mutual induction between circuits as indicated above so as to bring the two waves nearly into coincidence in length, but in general the existence of two wave lengths interfered with the sharp tuning which is the second requisite set forth by Sir William Crookes as being necessary to prevent interference in receiving.

Further investigation showed that after the oscillations started in the closed circuit by the discharge its stored energy through the spark gap had built up oscillations in the open circuit to a maximum the closed circuit momentarily came to rest. Then an immediate return to the closed circuit of the energy not already radiated from the open circuit commenced, setting the closed circuit again into oscillation. This retransfer of the remaining energy was complete after several vibrations and the open circuit momentarily came to a stop, only to be set in vibration again by a re-retransfer of energy, the two circuits passing the energy from one to the other until it was entirely dissipated either in heat or in electric waves.

It is evident that for any one discharge of the condenser if the total energy of this discharge is transferred to the open circuit, then any retransfer to the closed circuit is a loss, since losses take place at each oscillation of either circuit, and the oscillations of the closed circuit, while they produce electric waves, do not produce them efficiently.

In 1906 M. Wien observed that while using a spark gap made of metal surfaces placed very close together his measur-

ing instruments only indicated one wave in the radiating circuit—the aerial, as we call it.

A further investigation of this phenomenon appeared to indicate that the gap, possibly on account of the cooling effect of the large surface and mass of material used, regained its dielectric strength to such an extent after two or three sparks had passed that the remaining energy was not sufficient to break the gap down again, so that the closed circuit made but two or three oscillations. In other words, it struck the radiating circuit a blow or gave it a push, and then left it free to vibrate in its own period, and thus waves of but one length were radiated.

From what has been said previously about the manner in which the closed circuit transfers energy to the open circuit it appears that this form of spark gap, which is made up of a number of very short gaps between metal plates, and which has become known colloquially as the "quenched spark," prevents the retransfer of energy from the open to the closed circuit after the closed circuit has completed the first transfer and is momentarily stopped.

With the spark gap open it is no longer a closed circuit, and, therefore, whatever vibrations are set up are not in tune with those in the open circuit and they absorb little energy from it.

We, therefore, have two ways of generating electric waves of any desired length, namely, weakening the mutual induction between coupled circuits, and thus making the two waves generated almost coincide in length; and by means of the quenched spark throwing the closed circuit out of tune with the open circuit by opening the spark gap after the energy contained in each charge of the condenser has been transferred to the open circuit, leaving the latter free to vibrate in its own natural period.

Experiment has shown that aerials generating short wave lengths are more efficient radiators than those generating long ones. Experiment has also shown that short wave lengths suffer more absorption over land and sea than long ones. There is, therefore, for each wave length a limiting distance

of efficient transmission, this distance being that at which the greater efficiency of radiation as compared with a longer wave length is counteracted by the greater absorption.

For instance, in the Brant Rock experiments, the results of which are being published in the Bulletin of the Bureau of Standards, it was shown that a wave length of 300 meters is more efficient than a wave length of 600 meters up to distances of about 500 miles; that a 600-meter wave is more efficient than one of 1,000 meters up to a distance of approximately 800 miles; that a 1,000-meter wave is more efficient than a 600-meter wave up to a distance of approximately 1,300 miles, these effective distances being over salt water.

For over-land communication the greater radiation efficiency of the oscillations producing short wave lengths is counteracted at much shorter distances by the greater absorption of short waves. This absorption varies greatly with the character of the land surface—whether wet or dry, level or hilly or mountainous.

It had been previously demonstrated by Duddell & Taylor for the British Admiralty that for short distances the attenuation due to distance varies directly as the distance and does not fall off according to the law of inverse squares, as predicted by Sir William Crookes.

The loss due to absorption, which is an inverse function of the wave length, must, except at night, be added to the loss due to attenuation. The absorption losses at night are often practically zero.

Experiment has further shown that the distance which signals can be received from an aerial varies directly as its vertical height and directly as the amount of oscillating current in it. It also varies directly as the height of the receiving aerial, so that it may be stated that the efficiency of two similar aerials for wireless telegraph purposes varies as the square of their height. That is, if the height of both are doubled, the received current will be increased four times with a given current in the sending aerial.

Taking the lower limit of received current which will pro-

duce audible signals as 1/100,000 of an ampère, experiments indicate that in order to communicate 400 miles, with masts 200 feet high, using a 300-meter wave, we must have an oscillation current in the aerial of 1 ampère; to ensure good communication we should have 4 ampères.

If we wish to ensure good average communication at 2,500 miles with masts 450 feet high we would require with a 6,000-meter wave, a current of 50 ampères in the aerial to secure readable signals and about 200 ampères to ensure good communication.

If we could produce 4 ampères of oscillating current in an aeriel with an expenditure of 2 kilowatts, we could, with the same efficiency of transformation, produce 50 ampères with 25 kilowatts, and 200 ampères with 100 kilowatts. These figures are, as yet, only approximate.

The Marconi Company has established regular day and night communication between Clifden, Ireland, and Glace Bay, Nova Scotia, a distance of about 2,000 nautical miles, with masts about 230 feet high, using a wave length of about 7,500 meters. The electric plant which produces this communication is between four and five hundred horsepower at each station.

Marconi, as the result of his early experiments, stated that the action is directly proportional to the square of the length of one of the two conductors if the two are vertical, and of equal length and in simple inverse proportion to the distance between them.

He coupled this statement with another, which is as follows: "With parity of other conditions vertical wires 20 feet long are sufficient for communicating one mile; 40 feet, four miles; 80 feet, sixteen miles, etc."

The great improvement in apparatus which subsequently took place, coupled with the specific statement of the distances which could be obtained with certain heights of wires, lessened or rather obscured the value of this deduction, whose correctness recent experiments have verified, the "parity of other conditions" in Mr. Marconi's statement being evidently the same power used at each height.

The law as given by Marconi is not complete, as it failed to take into account the effects of absorption, which at the distances over which his experiments took place were not appreciable.

The recent Brant Rock experiments conducted under the direction of the Navy Department are the most extensive series of observations of this character whose results have yet been published.

It appears from the foregoing that the first requirement of Sir William Crookes has been fulfilled, namely, we have certain means of generating electric waves of any desired length. We cannot say that they are simple. In fact, they are more complicated than the methods which were familiar to him. In addition to knowing how to generate waves of any desired length we know, at least approximately, what wave lengths are best for given distances, and are in a fair way towards being able to design sending apparatus, including masts and aerials, whose radius of day communication we can guarantee with a given power expenditure.

Taking up his second requirement, "the invention of more delicate receivers which will respond to wave lengths between certain definite limits and be silent to all others."

He included in the term receivers, which he knew only as short resonators with which waves were detected by observing the sparks passing between their terminals, apparatus which we now divide into three classes: Receiving circuits, detectors, and means for making the action of the detectors audible or visible, such as telephones, ampliphones, Morse writers or recorders.

Referring to receiving circuits: Marconi in his experiments used as a receiver the apparatus shown in figure 5—two metal or tinfoil-covered glass plates connected by a tube containing metal filings, and in his patent application he described a means of tuning this receiver to the transmitter, stating that the length of the plates and conductors forming his receiver apparatus should be such "as to cause the circuit to resonate electrically in unison with the electrical oscillation transmitted

to it." He further states that he sometimes gives receiving apparatus the shape of an ordinary Hertz resonator with one of his sensitive tubes (coherers) in place of the spark gap. The coherer had been invented or discovered by Branly about 1890.

In completing his specification in 1897 Marconi describes a method of fixing the proper length of the plates of his receiving apparatus to make them resonate. In other words, he described his method of tuning. He did this by using plates of tin foil pasted on glass and making a cut across the foil with a very sharp penknife or point and bringing it close to the sending apparatus shown in figure 1 in such a position that the strips of tin foil are parallel to the line joining the centers of the oscillating spheres, when sparks would jump across the cut from one strip to the other.

When the lengths of the strips, which could be shortened or lengthened as required, were adjusted to approximate resonance the sparking occurred at the greatest distance from the oscillation producer, so that to tune his receiving apparatus he brought it to the sending apparatus and made his adjustments there, and then set it up elsewhere.

In the same specification he shows receiving apparatus illustrated in figure 6, which, it will be observed, is the exact reproduction of the sending apparatus shown in figure 2 with the detector in place of the spark gap.

When he used long, vertical wires as senders the method of tuning the receiving set by means of the sending set in its immediate vicinity was not practicable, and apparently no definite efforts were made to produce resonance between the sending and receiving circuits during the early experiments in England, for we read that in May, 1897, with the sending apparatus connected to a vertical wire 150 feet high the receiving apparatus was at first connected to a wire 90 feet high, about 60 feet above sea level, and for two days attempts to communicate over a distance of three and a-half miles were unsuccessful. On the third day the receiving apparatus was carried down to the beach, at sea level, and connected to the

90 feet of wire by the additional 60 feet necessary to reach sea level, making 150 feet in all, and communication was immediately established.

It was not, however, until 1899, two years later, that the great advantages of tuning the receiving set carefully to the sending set, and the further advantage of placing it in the closed circuit, as illustrated in figure 8, were appreciated by him.

This, it will be seen, is an inductive connection, and with such a connection considerable increase in the range of communication was immediately obtained. We think now that the cause of this increased range was due to the fact that removing the detector from the aerial and replacing it with an inductive coil of wire decreased the resistance of the open circuit and increased its self-induction, both of which effects tend to give it a more pronounced natural period, with smaller damping, and therefore enable it to respond to a succession of waves, each one adding to the strength of the preceding one, and thus produce a higher maximum of current in the open and also in the closed circuit containing the detector than could be produced with the detector directly in the open circuit. The resistance of nearly all detectors is very high for very low voltages.

At the same time that the direct-connected sending apparatus shown in figure 4 came into use direct-connected receiving apparatus was introduced, the connections being as shown in figure 7, except that the arrangements for variable connections in both inductance and condenser, and which are used for tuning both the closed and the open circuits, were not introduced at this time, but were a later development.

The arrangements illustrated in figures 6 and 7 are practically reproductions of figures 2 and 4, with the detector in place of the spark gap. Fig. 8 is the ordinary inductively-coupled receiving circuit.

While more direct coupled than inductively-coupled sending sets are in use in the world today, the inductive coupling is in the lead for receiving purposes.

Mr. John Stone Stone, of Boston, developed a method of sharp tuning by the introduction of an intermediate-tuned receiving circuit, which he introduced between the open circuit and the closed-detector circuit in a manner illustrated in figure 9. This intermediate circuit he called a "weeding-out" circuit. This is intended to prevent interference from static discharges and from electric waves of nearly the same length as those desired to be received.

Fessenden also produced an interference preventer shown diagrammatically in figure 10.

Both the weeding-out circuit as proposed by Stone (used also by Marconi for some purposes) and the Fessenden interference preventer are useful under certain conditions, but usually the inductively-connected tuned receiving circuit (Fig. 8), without any intermediate circuit, is that in general use.

Receiving circuits are not in general more delicate, as specified by Crookes, except in the sense of being more selective. They have a greater relative amount of self-induction (electromagnetic inertia), and are therefore less responsive to waves whose periods differ widely from the ones they are adjusted to receive.

Sending circuits are also less highly damped than was the case when the plain aerial was used, and more oscillations being contained in any one wave train enables the selectivity of the slowly-damped receiving circuit to be used to advantage.

Referring now to detectors: The coherer used by Marconi was improved in reliability and sensitiveness both by Marconi and by many other manufacturers, but not to any great extent after 1899.

For ordinary ship communication the reliable operating distance of coherers with expenditure of 2 to 3 kilowatts in the sending apparatus was from 50 to 75 miles. The signals received were read both by the sound of the tapper used to decohere the filings and from the tape printed by a Morse writer operated by a sensitive relay.

Many other kinds of detectors, such as various forms of microphonic and rectifying contacts were discovered, which could

be used with a telephone, but none of them proved to be sufficiently reliable to be placed in general use until the electrolytic detector was developed by Fessenden in 1902.

It was found to be so much more sensitive and reliable in action than any of the coherer detectors that the former were soon discarded in America for the electrolytic in various forms.

But Marconi in the meantime developed a magnetic detector, which was also suitable for use with the telephone, and which, though very reliable, has never compared favorably in sensitiveness with the electrolytic, but which was used to replace coherers.

The introduction of detectors such as the electrolytic, which could be used with a telephone, practically tripled the working range of telegraph sets.

In 1905 General Dunwoody, U. S. A., introduced carborundum crystal detectors. This resulted in the search for, and discovery of, a great many other crystal detectors, many of which are more sensitive than the carborundum and which rival and in some cases slightly exceed the electrolytic in sensitiveness, but which are less constant in their action.

The latter is specified as the standard for comparison when detectors are being purchased under contract by the Navy Department, the specifications providing that all detectors shall be equal in sensitiveness to those of the electrolytic type, using an anode of unjacketed platinum wire of .0002 of an inch in diameter, in a solution of 20 per cent. nitric acid.

The electrolytic detector requires the use of a local battery with positive pole connected to the fine platinum wire of the cell.

Some of the crystal detectors are slightly more sensitive if connected up in this manner with a local battery, but, generally speaking, will operate without it.

They rectify the received oscillating current so that it can be heard in a telephone.

Certain of the crystal detectors have a further advantage over the electrolytic of not being injured, as is the fine wire,

by static discharges or induced currents from the sending apparatus.

Crystal detectors are now in general use.

Special telephones have been developed for use with detectors.

The ordinary commercial telephone has a resistance of about 80 ohms in the receiver magnet windings. It was found that owing to the high resistance of the detector and the exceedingly small currents in the receiving circuit that the strength of signals can be increased by increasing the number of turns, and accordingly the resistance of the receiver magnets for wireless telegraph use.

These windings have been gradually increased in length and fineness of wire until the most sensitive telephones have a resistance of about 2,000 ohms.

It was also found that the best position of the diaphragm of the receiving telephone for reproducing speech is not the best for making audible the exceedingly weak impulses produced by the change of current in the receiver magnets. Telephone diaphragms have, therefore, been made adjustable, in order that the distance at which the diaphragm is placed can be regulated and the most sensitive point determined by actual experiment in use.

Recently it has been found possible to make the received currents work a microphonic relay or series of relays, which steps up the current to such an extent as to produce a comparatively loud signal, that is, one which can be heard in the operating room at some distance from the loud-speaking telephone used with the apparatus. These instruments are known as ampliphones or resonance relays, and have been made to operate polarized relays and Morse writers.

They give great promise of further increasing the range of communication with a given power, of producing a call which will decrease the number of operators necessary at a station, and of increasing the relative difference between two signals of nearly the same strength so that either one can be read as desired.

A consideration of the above statements relative to receiving apparatus will show that the requirements of Crookes as regards delicacy of receiving instruments have been entirely fulfilled, and that we have receiving circuits which, with some qualifications, "will respond to wave lengths between certain definite limits and be silent to all others," and we hope soon to have receiving apparatus which will only respond, first, to certain wave lengths, and, second, when the wave trains made up of these wave lengths are of a certain frequency.

We have now to consider the third requirement—the means of darting the sheaf of rays in any required direction. This has not been satisfactorily accomplished except for very short distances.

As is well known, Marconi generated free electric waves in the focus of parabolic mirror and directed them towards his receiver, which was placed in the focal line of the mirror, but the distances that he was able to cover by this method were so short that he soon discarded this for the vertical earthed-wire transmitter, which generated earthed waves.

Surrounding this vertical wire with a reflector by which the waves could be directed was manifestly impracticable, and no serious efforts in this direction were made for a number of years, because, though the vertical aerial did radiate waves in all directions, the distances over which communication could be carried on were so greatly increased that no return to the generation of free waves directed by mirrors was contemplated. The discussion of this feature of wireless telegraphy will be of attempts rather than achievements.

Many experiments in this direction have been made, notably by Artom in Italy; Braun in Germany; Guarini in Belgium; Belini & Tosi in France; Marconi in England; and Stone in the United States.

Artom attempted to generate elliptically-shaped waves by the introduction of multiple aerials.

Braun's experiments were along the same general lines, namely, to decrease the radiation in one direction and increase it in another.

Guarini attempted actual direction by means of refracting the waves. He proposed, as did Ladd, of Boston, to encase an aerial in a reflecting tube having a slit on one side through which directed waves would be emitted and the direction of received waves determined.

Marconi discovered that a transmitter having a large portion of the aerial horizontal, received more strongly in the direction away from which the horizontal wire pointed and sent more strongly in the direction from which it received the best. This was demonstrated by experiments with a British man-of-war.

Horizontal aerials radiating from a common center like spokes of a wheel were set up on shore in such a way that each one could be connected at will to a common vertical aerial. It was found possible to determine the direction of the sending vessel from the station on shore by connecting the horizontal parts in succession to the vertical parts. The strongest signals were received when the vertical part was connected to that spoke of the wheel pointing away from the direction of the sending vessel.

The trans-Atlantic wireless stations of the Marconi Co., at Clifden, Ireland, and Glace Bay, Nova Scotia, have the horizontal portions of their aerials, which are very long as compared with the vertical portions, pointed directly away from each other; but whether the sending efficiency is increased by these directive aerials in the direction opposite to that of the horizontal portion, or whether it is only decreased in other directions, has not been definitely shown. This method of directive sending and receiving is only applicable to shore stations. No satisfactory explanation of the cause of the directive effect has yet been offered.

Stone, in the United States, based his attempts at directive receiving on the fact that two vertical wires in a plane at right angles to the direction of the waves would have oscillating currents of the same phase and strength induced in them, which could be made to neutralize each other and produce zero signals.

If the plane of the wires coincided in direction with that of the wires and if they were half a wave length apart, the induced currents were opposite in phase and could be used to augment the strength of signals so that signals could be made a maximum in one case and a minimum in the other. A revolution of the ship in azimuth carrying two such vertical wires would cause the strength of signals to vary to such a degree that the direction of maximum or minimum signals could be determined, and thus the plane of the passing waves.

Experiments with this method were made between the *Lebanon* and a shore station in 1906, and were moderately successful.

An attempt was subsequently made to determine the relative bearing of two ships from each other by this method, but the difficulties connected with balancing two aerials so as to produce zero signals were found to be too great to permit making practical use of this system.

Ladd, in Boston, and Bellini and Tosi, in France, showed that aerials installed as above received more strongly when their plane coincided with the direction of the passing waves, regardless of their distance apart relative to the wave length, and if revolved rapidly in azimuth, as can be done if supported by a mast on shore, were excellent direction indicators. Bellini and Tosi have applied this to ships where actual swinging of the vessel is undesirable by installing vertical wires forming two planes at right angles to each other. The receiving apparatus is arranged so that the plane of the receiving coils relative to the planes of the double aerials is known, and its position when the strength of signals is a maximum, indicates within limits of one or more points, the plane of direction of the passing waves. This apparatus is in practical use to a limited extent.

The importance of having a reliable means of determining the direction from which wireless telegraph signals are coming cannot be overestimated. Being able to direct them is of equal importance. The Bellini and Tosi apparatus is suitable for

receiving and determining the plane of the waves—not their actual direction in that plane.

Referring to a further statement of Sir William Crookes that, "considering there would be the whole range of waves to choose from, varying from a few feet to several thousand miles":—He believed that "there would be sufficient secrecy, for the most inveterate curiosity would surely recoil from the task of passing in review all the millions of possible wave lengths on the remote chance of ultimately hitting on the particular wave length employed by those whose correspondence it was wished to tap." However, practical considerations have limited the number of wave lengths in use, and receiving circuits have not proved to be in general so highly selective as to respond only to wave lengths within a few per cent. of those to which they are adjusted.

There is sufficient range of possible wave lengths to provide any needful communication if fixed stations only were necessary, but since the principal use and development of wireless telegraphy has been with and between ships at sea, which are continually moving and varying their distances from each other and from shore stations, the problem is by no means a simple one. Except for special stations, like those at Clifden and Glace Bay, which are designed to communicate only with each other, both ship and shore stations are almost forced to a common wave length in order to be mutually ready to receive from each other and any other station; and a sharply-tuned receiver is as great a disadvantage for some purposes as is a broadly-tuned one.

A station that must be ready to hear any number of vessels can only hear them all if all send with the same or nearly the same wave length, and this applies also to vessels desiring to communicate with each other. If vessels in the vicinity of a shore station desire to communicate with each other and at the same time other vessels desire to communicate with the shore station, interference cannot well be avoided. Practically the only recourse is to a division of time.

One operator can be ready to receive two wave lengths at

the same time, making use of two receiving sets, one connected to each ear; but this does not cover the case, and, generally speaking, the custom is to have the receiving circuits broadly tuned for listening in, and then adjust them as sharply as may be to non-interference if several communications are being carried on in the same vicinity in the common wave lengths.

The arrangements for shifting from board to sharp tuning and from one wave length to another for sending and receiving are not as simple as the statement of Sir William Crookes might lead one to suppose, though it is "accomplished by turning a screw or altering the length of a wire." The process usually involves several adjustments of capacity, self-induction and mutual induction, and it is only now that mechanical arrangements are being perfected for altering the wave lengths throughout the entire range of apparatus, both for sending and receiving, by the manipulation of a single handle for each purpose, and at the same time maintaining resonance.

The general introduction of apparatus of this character will enable greater advantage to be taken of the possibility of assigning other non-interfering wave lengths for communications which need to be carried on over the same area at the same time.

The very long wave lengths, such as those used by the trans-Atlantic stations, do not affect ordinary receiving apparatus tuned to the shorter waves and cannot be read unless special loading coils for ships' sets are provided to increase their range of receiving.

The importance of the subject of wireless telegraphy is indicated by the fact that it has been the subject of two international conferences, and that a third is to convene in London, either in 1911 or 1912.

A law has been passed, which goes into effect on July 1, 1911, requiring any ocean-going steamer of the United States plying between ports over 200 miles apart, and any foreign steamer sailing from a United States port, carrying 50 or more persons, to be equipped with an efficient wireless-tele-

graph set capable of transmitting and receiving messages at least 100 miles day or night.

The second International Conference, held in Berlin in 1906, established wave lengths of 300 and 600 meters for shore stations for general public use; provided for the use of any wave length between 300 and 600 meters for ship stations, with 300 meters as a standard for communication between ships and shore stations. Wave lengths between 600 and 1,600 meters were reserved for naval and other Government stations, and wave lengths greater than 1,600 meters were reserved for special shore stations, each station to be assigned a particular wave length, as might be directed.

Congress has not yet ratified the recommendations of this conference and the United States is, therefore, not a party to the agreement, though it participated in both conferences.

Wireless-telegraph companies in the United States have been placed under the jurisdiction of the Interstate Commerce Commission.

In the principal foreign countries wireless-telegraph stations are operated or licensed by the general government.

In the United States the operation of stations by commercial companies has been under charters taken out under the laws of various States, and they have been without supervision as far as regards interference with each other and with Government stations.

Wireless telegraphy is indispensable to the efficient operation of navies, both in peace and war.

Consideration of what has been said in this article will show that we have realized nearly all of Crookes' predictions, though much remains to be accomplished in the way of accurate measurements of sending and receiving currents under different conditions and in perfecting mechanical construction, in order to put the design of wireless-telegraph sets on the same plane as that of apparatus for other purposes.

Although we have not yet reached the last stage in the imaginings of Sir William Crookes when two friends having their

instruments attuned to their desired wave lengths could call each other at any time or place and communicate at will, still we are able to say that Job, if he had lived in these days, might have been comforted by being able to answer "Yea, Lord," when asked "Canst thou send lightnings?" "that they may come unto thee and say:—Here we are."

NOTES ON THE BURNING OF LIQUID FUEL.*

By Luther D. Lovekin, Associate.

The efficient burning of liquid fuel for the generation of steam is a matter of world-wide importance. On steam vessels especially, the advantage of oil over coal, as a fuel, are most important, and lie not only in an increased mileage per ton of fuel carried and in the ready possibility of smokeless combustion, but also, and mainly, in superior elasticity in furnace control, whereby almost instant response can be had to sudden demands for greatly increased power and speed.

It may therefore be easily understood why oil fuel has been specified for late naval vessels and particularly for vessels of the "Destroyer" class, which develop abnormally high power for their displacement, and which must necessarily possess the characteristics above noted in the fullest obtainable degree.

It is a well established fact that there is at present only one way to burn liquid fuel for steam raising, and that is by atomizing the fuel in company with a sufficient amount of air around each atom. In order that oil may atomize freely, it should be deprived of its viscidity. This is really done by heat, and brings with it the further advantage that any water in the oil more easily separates from the hot oil than from cold oil. First, because the heated oil, being more limpid, offers less resistance to the freeing of the water, and, secondly, there is a greater expansion of oil than of water, due to the heat, and the water gains a relatively greater specific gravity.

With oil a maximum fire is almost instantaneous; the only drawback is the cold brick work of the setting, which must

*A lecture delivered at the School of Marine Engineering, Annapolis, Md.

become hot before the maximum furnace duty is obtained. For ordinary economical work the number of heat units that a boiler can absorb per square foot of heating surface will not be changed when liquid fuel is employed, excepting smokeless combustion being far more practicable with fuel oil than with the usual steaming coals, these surfaces can be maintained in a cleaner and therefore more efficient condition.

As with bituminous coal, which, like oil, is a complex hydrocarbon, liquid fuel should be burned in furnaces more or less protected from immediate loss of heat to the boiler surfaces by means of lining or baffles of fire-brick. Liquid fuel, however, is more easy to burn completely than coal, because it can be mixed more intimately with the necessary air. The interior of a combustion chamber should show a clear white incandescence with little apparent flame and with no smoke nor unburned gases coming from the chimney. If looked at through a piece of violet-colored glass, the interior of the combustion chamber with its brick linings should show a light lavender color, indicative of perfect combustion, with the production of actinic rays showing probably intense chemical action. A chilled fire, such as is produced in boiler furnaces with a coal fire, will show very little light indeed through a violet glass, its flame being cut down, in many cases, from several feet in length to a few inches. The flames of yellow and reddish intensity being resolved into rays through dun-colored glass, will throw off no light of sufficient actinic power to penetrate the glass. The whole question of burning oil is really a simple one, and resolves itself into exactly the same condition as does the coal question, namely: an intimate mixture with air in sufficient quantity and a proper conservation of the temperature pending full combustion.

A minimum of 15 pounds of air is necessary to supply the oxygen for combustion of each pound of average hydrocarbon liquids, but probably at least 5 per cent. to 10 per cent. excess is required in the very best practice. Statements have been made that from 50 per cent. to 100 per cent. excess of air is necessary, depending on the system used and the condition

of the furnace in which it is burned. It has been stated that in order to insure smokeless combustion of liquid fuel not more than from 2 to 3 pounds should be consumed per hour per cubic foot of combustion space. Our present destroyers have, however, 1,747 cubic feet of combustion space in four boilers, and are required to burn about 17,000 pounds of oil per hour, or nearly 10 pounds per cubic foot of space.

Fuel oil will not burn without spraying when cold, and cannot be ignited in a mass. If heated to the point of ebullition and supplied with air it will burn freely. The art of burning liquid fuel consists in heating only the portion which is to be immediately burned, and exposing it to contact with the air. Unlike coal, it is not possible to burn it at many surfaces. A coal fire is made up of many pieces of coal, each burning over its whole surface. With oil fuel we arrive in practice at a system in which we employ a jet or spray in order to split up the stream of the liquid into fine globules in the presence of a sufficient supply of air. Each globule burns superficially and becomes heated by its own combustion and the general heat of the furnace; and this principle appears to be the best and most effective method of burning liquid fuel—in fact, it is the best method of burning anything—first to reduce it into particles so fine that their bulk bears a small ratio to their surface area, whereby each particle is brought close to the air which it requires. The necessity for atomizing arises purely from the insufficient area of the fuel otherwise treated. A fire composed of lumps of coal is full of interstices, and the area of the fuel exposed to the air is much greater than the area of the fire grate; hence if the fire is to be controlled the fuel must be supplied as it is consumed, so that at no time is there any serious amount of burning fuel in the furnace, and the production of steam is at once regulated by the amount of the fuel and air supplied. This end is secured by atomizing the fuel and discharging it into the furnace so mixed with air that each atom of fuel is in contact with air, and combustion easily effected. It will be found that with all the heavy liquid fuels that atomizing is essential.

Oil fuel will produce dense smoke when not supplied with sufficient air, but in all approved methods of combustion the requisite air is supplied and can be regulated very exactly.

All smoke troubles may be attributed in general terms to the too early application of the heat-absorbing surfaces of the boiler to the yet unconsumed gas. With liquid fuel the supply of gas is uniform and continuous, and the fuel is supplied in exceedingly small particles intimately mixed with air to begin with, and sometimes with a further volume of air from below.

A uniform high temperature is maintained in the locus of combustion by a sufficient mass of fire-brick work in the form of arches or chequer work, or, as in the case of the destroyers, by means of circular tiles of fire clay surrounding the oil spray, which become incandescent. In addition to these we provide brick work extending over the entire bottom, sides, back and front of the boiler. It may be added that the length of flame from a burning hydrocarbon is largely determined by the intensity of the combustion as well as by the perfection of the air admixture. A well mixed gas, burning at a high temperature, will produce a short flame, whereas the same gas burned in a water-cooled boiler will produce an exceedingly long flame. In the ordinary form of water-tube boilers, where gases rise directly between the water tubes, the proportion of cold surface at once encountered by them is very great, and while it is quite possible to arrange this type of boiler so as to burn oil fairly well, its construction prevents high efficiency being obtained.

The proportion of rays of any particular color in a furnace will indicate the intensity of the action which is going on within the furnace. It is extremely difficult for the most highly experienced eye to discern the full action of a furnace at high temperature, not perhaps so much because of inability to estimate the relative value of color present, but because of the superabundance of heat rays which accompany the chemical rays and the dazzling effect of even moderate temperature. Hydrogen ignites so very much more readily than carbon and

at so low a temperature that the probability is that hydrogen does burn, and in doing so takes the available oxygen from the surrounding air and deprives the adjacent carbon of any opportunity of combustion, causing it to deposit as soot, so that unless there is sufficient temperature there is no hope of burning oil without the formation of smoke. Temperature is so closely connected with actinism that the analytical investigation of the light of a furnace will give a fair insight into its conditions of temperature.

By means of transparent media of suitable composition light may be analyzed in a manner that will afford great assistance in arriving at sound conclusions and practice. Such media are colored glasses. A ruby-colored glass will cut off all rays of light of higher vibration than ruby color. Only the lower end of the spectrum will be visible through such a glass. On the other hand, by means of a violet-colored glass, all the less active rays than violet will be eliminated, and the most brilliant of furnaces may be thereby rendered easily visible, its interior being colored the peculiar lavender-gray color or approaching this tint, which marks the ultra-violet end of the spectrum. The more perfect the combustion the larger will be the proportion of violet light emitted by the flame. Occasionally in the mass of flame dark streaks may be seen; these represent streams of burning gas, which, while incandescent, are below the violet stage. They may be traced to a point of disappearance, and they would probably radiate some light if the color of the glass were less violet and more blue.

The effect of radiation is to cool the flames below the range of violet long before they have risen to the level of the tubes, and apparently there is nothing but radiation to explain the reduction of temperature. This method of analysis of the products of the fire is useful, not merely because it enables a furnace interior to be visually examined with ease and comfort, but because it shows so clearly the effect of a good design and the bad influence of premature cooling. It affords conclusive testimony to the benefits that accrue from proper designs.

The use of violet-colored glass is essential, not simply because it reduces the amount of light which meets the eye and renders vision possible, for such a result could be obtained by smoked glass; but the violet glasses or the higher blue colors, are necessary because of their especial analytical properties. As it is very probable that the violet degree of actinism must be fully developed if combustion is to be perfect, and this degree of actinic effect cannot be associated with temperatures that can be secured in any furnace so arranged that the gases rise vertically from the grate surface to pass between the water tubes before they have been thoroughly commingled and burned in a free space; thus the ordinary design of a water-tube boiler is a very inefficient one for oil burning, and the greatest attention should be given to the furnace design of a boiler of this type in order to secure a refractory lining throughout all its parts. This, however, must be limited so as not to complicate the construction or prevent access to the important parts of the boiler.

Feeding the oil at a temperature approaching the point of distillation insures speedy vaporization with a resultant flame soft and diffusing, and not sharply impinging on the boiler surfaces. It is also true that oil atomizes with less expenditure of energy when heated to such temperature.

The temperature of the escaping gases from a boiler will increase as the excess of air becomes greater, provided the same amount of fuel is being burned. This is because the furnace temperature is less, owing to the greater amount of air, which results in a less rapid transfer of heat through the boiler, and consequently allows more heat to escape through the chimney. On the other hand, with a uniform excess of air, if more fuel is burned the temperature of the escaping gases will increase, owing to the heat produced being greater in proportion to the absorbing capacity of the boiler. The gases should not leave the boiler at too high a temperature, and this can be controlled and regulated largely by the aid of air regulators. A clear flame consumes less oil than a smoky flame, and has a greater efficiency. Smoke is an evidence of

imperfect combustion, but the absence of smoke does not necessarily demonstrate or prove that perfect combustion is being attained.

The flame of oil is usually a long one, and care should be taken to suitably arrange the burners so that the flame may not pass between the tubes and become extinguished, causing the gases which are partly burned to relight in the chimney.

The chemical action and reaction of a burning spray of oil may be very much complicated by disassociation or even by exothermic formation, which may delay heat production; later, when combustion becomes active, as shown by the light-giving power of the flames, it will be more or less rapid, according to the perfection of air admixture, and will last for a time (t), during which the jet, traveling at a high velocity (v), passes through a distance $D = v\ t$, which may be yards in length; thus the course of the gas must be long or it may escape too hot to the chimney. Hence arises the necessity of cutting short the flame by an early admixture and a high temperature, so as not to lose the benefit of the boiler-heating surface. It is for this purpose that in most successful oil-burning furnaces the jet of atomized oil is directed upon a brick obstruction of some kind, so as to spread the flames and cause them to fill the furnace space.

To burn the oil properly and successfully it must be divided into very fine particles or atomized, and should be heated sufficiently to destroy its tenacity, and should be dispersed into a fine spray intimately mixed with air in proper proportion.

Recent trials with liquid fuel show that there is still much to be done in reducing the air supply. Ordinary oil fuel requires fully 15 pounds of air per pound of fuel, with, of course, some additional excess as with solid fuel, but with oil fuel there should be a better mixture of air and fuel than when coal is burned, and therefore better combustion with less excess of air.

The large amount of air per pound of oil arises from the large percentage of hydrogen in the oil, and it is the great capacity for oxygen possessed by hydrogen which renders the

theoretical temperature of combustion so nearly like that of carbon, in spite of the high calorific capacity of hydrogen.

As regards the safety of oil, it has been shown that oil having a flash-point of 239 degrees F. would not ignite if fired into with a shell, nor would dynamite exploded in a reservoir of this oil do more than throw up jets of oil which would not ignite. Any danger with liquid fuel is practically confined to those oils which have not parted with their inflammable and volatile gases, and this is a danger with oils when used absolutely crude. When these portions are removed, however, as is the case with all ordinary commercial fuel oils, oil is safe, and, moreover, unlike coal, it contains no power of spontaneous combustion.*

Various devices have been made for the purpose of securing a complete atomization of the oil, and many are in use at the present day; but, after all, it is a very simple matter now to make an oil burner that will atomize, and do it well; but even in the case of atomizing there is a great difference between doing it well and doing it almost perfectly.

FUEL-OIL INSTALLATIONS ON DESTROYERS.

While the following applies more particularly to the destroyers *McCall* and *Burrows*, the experience on these vessels has developed data which are believed to be of application to boiler fuel-oil installations on this class of vessels in general.

When the question of burning oil for our new destroyers first came up I was invited to prepare plans of a system which I was interested in developing; but, after due consideration, it was found that on account of the apparatus (which was known as the Lassoe-Lovekin System, using low-pressure air) being so heavy, it was not adaptable, and, in fact, entirely unsuitable for this particular work.

Atomizing by steam being out of the question for marine

*The preceding extracts, most of which have been taken from the excellent work of W. H. Booth, entitled, "Liquid Fuel and its Combustion," have been of such great value to me in analyzing this subject of oil burning in actual practice, that I feel it my duty to give the author due credit, and to recommend a complete perusal of his work.

purposes, owing to the loss of fresh water involved, attention was turned to the direct-pressure system, and after numerous inquiries it was found that Dr. Koerting had invented a process for the burning of liquid fuel which looked very sound. After a conference with our officials (New York Shipbuilding Co.), we decided to place our contract with the Schutte & Koerting Company for supplying the outfit.

We had no opportunity for making any tests which would be of value, with the exception of a relative test on spraying, which was done and which satisfied us that, so far as the spraying of the oil was concerned, no one could ask for better results.

Accordingly, this system was installed on the *McCall*, and when ready for trial the vessel proceeded to the Delaware Capes, and, to the surprise of everyone, its later complete success was not at this time demonstrated.

Vibrations and flare backs shook the furnace bricks down and caused general alarm among all concerned. Several difficulties existed, the principal one was being unable to produce a steady oil pressure, so air vessels were installed, charged with air on the dead ends of the oil-delivery pipes, as well as alongside of the oil-pump, and these had the effect of producing an absolutely steady oil pressure at all times.

Having solved the steady oil-pressure problem, it was decided to try again, and, to the surprise of all, we still had troubles, and the question was to find out what the difficulties really were. This proved quite puzzling at first, but after giving the subject much study, I arrived at the conviction that in order to burn oil properly we must have a constant relative supply of oil and air, and that this mixture should enter the furnace "arm in arm" in its respective proportions, so as to form a perfect gas, which, if possible, would enter the furnace in form for perfectly quiet combustion.

I made calculations of the velocity of oil and corresponding calculations on air, which proved to me that there was but one direction in which to work, and that was to increase the velocity of the air without increasing the volume; so, we

closed the air-registers to a sufficient extent to give the proper amount of air at its proper velocity, and immediately all previous difficulties disappeared.

The *McCall,* together with her sister ship, the *Burrows,* has now run all of the Government trials without a single mishap, vibration or flare back.

To facilitate the work of investigation curves were plotted from tests of the blowers, which were of the greatest possible value and which will be equally valuable to those charged with running these vessels in service, as from these curves it is possible to readily determine the amount of air required at various blower speeds and air pressures. Another important curve was plotted, from which the amount of oil that can be forced through the type of oil burner used under various pressures could be readily ascertained. These two sets of curves are of the greatest possible importance to those who may be called upon to operate these installations, and, in fact, similar curves, if not already furnished, should be provided for all vessels of this class, as they enable the engineer to know all the possibilities at his command, so that intelligent experiments may be carried out and thus the best economy secured under all conditions.

These oil-pressure curves can very readily be compared from time to time with the actual amount of oil used within a given time and corrections made accordingly, or a known percentage can be used in order to obtain the actual results.

The exact velocity of air required for any oil-burning system of the direct-pressure atomizing type is indeterminable in advance and must be the result of experiment, for the reason that it depends on the volume of the furnace, the resistance of the uptake, the resistance through or between the tubes, and the resistance through the smoke pipe. Furthermore, it depends on the dilution of air used, as the volume of gases depends on the temperature of the furnace, while the temperature of the furnace depends on the air dilution. There is every reason why we should aim for a small dilution of air, and the nearer the theoretical requirements can be met the

higher our furnace temperature will be. This shows conclusively that all oil-burning outfits should have either an adjustable air supply with a fixed oil supply, or a fixed air register with an adjustable oil burner, or a combination where both air and oil supply are capable of being adjusted.

The location of the tip of the oil burner is best found by experiment, although it can be arrived at very closely by experience.

Inasmuch as the spray from an oil burner varies in intensity from its tip outward, and the strength of the jet of oil is under pressure of from 100 to 200 pounds any thought of penetrating this jet with air of only 5 inches of water pressure must be dismissed, since it is impossible to accomplish this result under such conditions. What we must do, however, is to surround this jet or spray of oil with air under sufficient pressure to give it a constant relative velocity to that of the oil spray, so that each atom of oil takes up its full quota of air and enters the furnace as a gas ready for proper combustion. The same principle is practically what is carried out by the carbureter of a gas engine.

In order to determine the effect of a high velocity of air, which was produced by closing down the air registers around oil burners, we carried the air pressure in the after firerooms during the five high-speed runs (official standardization trials of *McCall*) at a mean of 6.35 inches of water. This required the blowers being run at an average of 1,460 revolutions. The boiler rooms served as large air reservoirs, so that the velocity through the air registers could be kept constant without any attention whatever from the fireroom force. The space between the oil spray and the air registers was of small volume, and therefore the drop in pressure was negligible; likewise, the effect of entering and leaving the fireroom through the air locks was unnoticeable, as was also a variation in the revolutions of one fan as compared with the other in the same compartment. No surging whatever took place and no vibrations or explosions during any of the runs.

The effect of the 6.35 inches of air pressure in the fireroom

was to produce a velocity of about 10,000 feet per minute through the air registers, which was considered would be a maximum for any speed we would aim for, it being previously understood that 32 knots was to be the maximum.

From the curves of the performances on these fans made by the Sturtevant Company's engineer, Mr. Williams, which are believed reliable and accurate, according to Mr. Taylor's method of measuring air, we could readily see the effect of a high air pressure affording a much less quantity of air at a fixed number of revolutions than a lower air pressure at the same revolutions. We had originally figured on 20,000 cubic feet of air as the maximum we would require at 5 inches air pressure.

We had previously made tests on the amount of oil that could be forced through an oil burner of the type which we are using, and which had a bore of 2.3 millimeters. These tests were each run for half-hourly periods, with readings taken every five minutes, and if variations in readings were found the entire series of tests were discarded. This required time and patience, and was finally accomplished by using a triplex pump, driven by an electric motor, so as to avoid any fluctuation in oil pressure. The curve on Plate I shows the result of these tests. Oil gages were calibrated both before and after these trials and the two on after boilers were found correct.

With this information at our disposal, both in regard to air and oil characteristics, we were enabled to investigate the results of this trial with some degree of intelligence; therefore an analysis of conditions with the two after boilers only, which we kept steady and which had all twenty-two burners in use, should prove of interest.

DATA FROM STANDARDIZATION TRIALS.

Oil gages absolutely steady throughout trials.	
Air pressure, average, five runs, inches of water	6.35
Oil pressure, average, five runs, pounds	180
Revolutions of blower, average, five runs, per minute	1,460
Cubic feet of air from curve, average of five runs, per minute, each blower	16,100

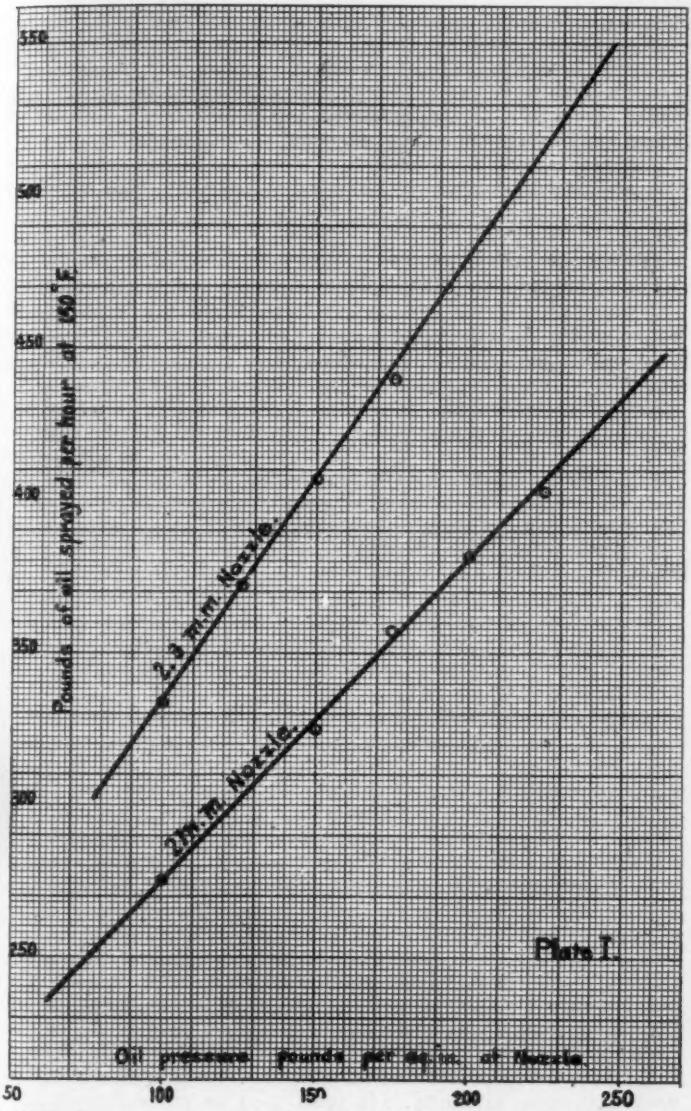


Plate I.—CURVES SHOWING CAPACITY OF "KOERTING" TYPE OF OIL BURNER WITH OPENINGS OF 2 AND 2.3 MILLIMETERS RESPECTIVELY.

Capacity of burner in pounds per hour at 180 pounds pressure, pounds	454
Number of burners in use in after firerooms	22
Temperature of oil at burners, average, degrees F	225
Size of bore in tip of burners, millimeters	2.3

Theoretical amount of air required per pound of oil is supposed to be 15 pounds, or 195 cubic feet.

Note.—Boilers were not thoroughly heated up when this trial started, due to a wait of about one and one-half hours for suitable tide conditions.

Two blowers in one fireroom, cubic feet	32,200
Total possible oil burned 22×454, pounds	9,988
Less 6½ per cent. corrected by actual tests, per hour	9,340
Heating surface, two boilers, square feet	9,600
Cubic feet of air per pound of oil, cubic feet, $\frac{32,200 \times 60}{9,340} =$	205
Air inlets by test showed a total of, cubic feet	215

During these runs the smokepipe temperatures were very high, and in fact the temperatures in the furnaces were so high that it was impossible for the boilers to abstract the amount of heat passing through the same, which accounts for the low rate of water evaporated per pound of oil. This, however, is not unusual with vessels of this class, where economy at maximum speed is of secondary importance.

It was very evident at the end of these trials that we were running with a very small dilution of air, as a dull red flame appeared at the top of boiler D stack for a few minutes during the last run over the course. This indicated a lack of air supply, but the trial was over and no further opportunity existed for further experimenting under this maximum condition.

It must be noted that we were burning oil at a rate slightly less than the British Admiralty requirements, which limits the rate to one pound of oil per square foot of heating surface, and which from our experience seems the limit of oil burning with this type of boiler. It must also be remembered that these vessels are intended to make a speed of 29½ knots, and

that the boilers were designed to be suitable for a maximum speed of 31 knots; the after boilers were making on the five high runs sufficient steam for a speed of 33 knots, if all four boilers had been forced to the same degree.

During these standardization runs the two forward boilers were used to keep up steam to a predetermined point and yet prevent blowing off, orders having been given not to exceed a certain gage pressure on the main turbines, consequently, only nineteen burners were used on these two boilers, and these were kept at an average of 168 pounds corrected oil pressure. The gages read 178 and 185 respectively, but were found after trial to be incorrect by calibration with standard mercury gage. The fact that the gages on the two after boilers remained correct by calibration after trial enables us to place reliance in the data.

By curve of capacity of oil burners, we find as follows:

Oil pressure of 168 pounds from curve gives 435 pounds per hour. $435 \times 19 = 8{,}265$ pounds per hour.

After boilers were burning 9,988 pounds of oil at the same time, or a total of 18,253 pounds. Difference between capacity by curve and actual measurement made by trial board on other trials of this vessel was found to be about $6\frac{1}{2}$ per cent. less, which would make 17,067 pounds of oil per hour.

Steam evaporated could not be ascertained on this trial, although we had an average B.H.P. of 14,333, and we know the water rate must have been approximately the same as that obtained on the four-hour run. Therefore, assume 14.2 pounds per B.H.P.

$$14{,}333 \times 14.2 = 203{,}528 \text{ pounds of water.}$$

$$\frac{203{,}528}{17{,}067} = 11.9 \text{ pounds of water per pound of oil.}$$

This, as before stated, was when the boilers were being forced at an extravagant rate and not intended to be economical.

DATA FROM FOUR-HOUR TRIAL—U. S. T. B. "M'CALL."

Four-hour trial steady conditions in after fireroom.

After boilers only considered, as forward boilers were varied to suit conditions.

Average revolutions per minute of blowers for four hours..... 1,411
air pressure in fireroom for four hours, inches..... 5.02
temperature of fireroom for four hours, degrees F..... 74
Number of oil burners in use for four hours..... 22
Size of opening in tip of burner, millimeters..... 2.3
Pressure of oil at burners, average for four hours, pounds..... 154
Capacity per hour of each burner at 154 pounds pressure, pounds.. 415
22 burners (22×415), pounds..... 9,130
Temperature of oil at burners, degrees F..... 220
Average temperature of smokepipes "C" and "D," degrees F..... 925
Less 7 per cent. as determined by trials—say, 8,500 pounds of oil per hour.

Boilers:

Heating surface of two boilers, square feet..... 9,600
Oil burned per square foot of H. S. $\left(\frac{8,500}{9,600}\right)$..... 0.885

Blowers:

Average revolutions per minute..... 1,411 } 18,600 cubic feet of air by
air pressure, inches..... 5.02 } "Taylor."
Capacity of two blowers, 37,200 cubic feet of air per minute.
Cubic feet of air per pound of oil $\left(\frac{37,200\times 60}{8,500}\right)$..... 263

We added some to the efficiency of the fans by cones in inlets. Probably not over 5 per cent. was gained in this way—say, 276 cubic feet, maximum, possible air used per pound of oil.

Four Boilers:

Average evaporation per hour, for four hours, was 185,647 pounds of water for all purposes.

Total oil used, 15,081 pounds per hour (measured).

$\frac{185,647}{15,081} = 12.31$ pounds of water per pound of oil, assuming 7.48 pounds of oil to the gallon.

No. of burners in use on after boilers..... 22
Oil pressure, pounds..... 160
Weight of oil used per hour (2.3 mm. bore) pounds..... 425
Air pressure, inches of water..... 5
Revolutions of blower, per minute..... 1,350
Temperature of oil..... 225
Amount of air delivered by each fan at 1,350 revolutions and 5-inch air pressure, cubic feet..... 16,400
Two fans, cubic feet per minute..... 32,800
Amount of oil burned by 22 burners at 160 pounds pressure, lbs.... 9,400
Less 7 per cent. for correction, pounds..... 8,742
Cubic feet of air per pound of oil $\frac{32,800\times 60}{8,742}$..... 225

ANALYSIS OF OIL USED.

Specific gravity at 50 degrees F.	.8902
55 degrees F.	.8882
60 degrees F.	.8865
72 degrees F.	.8820
Flash, degrees F.	300
Fire, degrees F.	320
Water and sediment	under 1 per cent.
Reaction	slightly acid.
B.T.U., per pound	19,618
Cold test, degrees F.	5
Ultimate analysis, sulphur, per cent	0.25
hydrogen, per cent	12.96
carbon	86.63

DESCRIPTION OF FUEL-OIL ARRANGEMENT ON TORPEDO-BOAT DESTROYERS "M'CALL" AND "BURROWS."

These destroyers are arranged with storage tanks for the fuel oil at the forward and after ends of the vessels, and with settling-tanks on either side of the boilers. The sides of the settling tanks are within about four inches of the boiler casing, while the top of the settling tanks is about one foot above the top of the lower drums at the sides of the boilers.

Those storage tanks in the after part of the vessel are divided into three distinct tanks, two of which are immediately adjoining the after engine-room bulkhead, while the third tank is separated from the other two tanks by the handling room, the storage of warheads, and 3-inch ammunition

These tanks receive their supply of fuel oil by means of a 4-inch deck-plate hose connection, and this 4-inch connection is divided below the deck into three branches, each branch being 3 inches in diameter and having a valve adjoining the T-piece for each branch. The valves are operated from the deck through separate deck plates.

These branches lead through to the top of the tank and connect with an automatic float valve which is located within the oil tank. The purpose of these automatic valves being fitted is to prevent the oil tanks from being filled to more than 95 per cent. of their capacity, thus allowing 5 per cent.

in the space for expansion. It is very important that these valves be of first-class manufacture, that they have conical seats and that the pressure of the oil in filling is on the top of the valve, so as to minimize the possibility of any leakage. These valves have tail pipes leading to the bottom of the oil tanks, and therefore the oil must drop through a distance of about eight feet.

Each compartment is provided with a 3½-inch air escape and overflow pipe with 3½-inch automatic valves above the deck. These valves are designed to admit air while pumping from the tank, and also to allow gas and air to escape while discharging to the tanks. They are fitted with wire mesh to prevent back-firing. The compartment is also provided with 1½-inch sounding tubes at their lowest points. These sounding tubes have a half-inch steam connection taken from a point near the deck casting for steaming out oil tanks. The extreme after compartment has a further ½-inch steam connection extending around the lower part of the tank in order to thoroughly steam out these tanks which are located under the crew's space and the chief petty officers' quarters.

Each tank is provided with two manhole plates for access. Too much care cannot be exercised in removing these covers in order to prevent the possibility of an explosion, and those in charge should be so instructed. While the manhole covers are removed, no smoking nor naked lights should be allowed in these rooms, which open directly into the crew's space and petty officers' quarters.

From each of these storage tanks there is a 4½-inch pipe, which runs to a manifold located in the engine room on the after end of the engine-room bulkhead. These valves are operated from the engine room as well as from the deck above.

From this manifold a 4½-inch suction pipe is led through the engine room and through both boiler rooms on top of the settling tanks to another manifold at the forward end of the boiler-room bulkhead. This forward manifold is for controlling the suction from the three forward oil tanks, each

of which is fitted with a 4½-inch pipe leading to said manifold. From this suction main there are two connections, one to the low-pressure pump in the after boiler room, and one to the low-pressure pump in the forward boiler room. These pumps are located on the starboard side near the shell plating just above the settling tanks. Each of these pumps draw through a 4½-inch "Macomb" strainer which is located immediately adjoining a 4½-inch stop valve on the pump. These strainers should have a finer mesh wire than is used for ordinary purposes.

The forward storage tanks are provided with 1½-inch sounding-tubes and 3-inch air pipes. Between the forward oil tanks and the forward boiler-room bulkhead is a cofferdam of one frame space in length. This cofferdam is also provided with a 3-inch automatic air valve. Both the forward tanks and the cofferdam are arranged with an ample number of manhole plates for access and for filling the compartments with oil, by means of a hose being led through the openings. (No hose connections are provided for the forward tanks.)

The settling tanks at the sides of the boilers are filled by means of the two low-pressure pumps previously mentioned, each of which discharges through 4-inch "Macomb" strainers, thence through a 4-inch pipe on the starboard side, with suitable valves arranged beyond the strainers so that either pump can supply any one or all four settling tanks by means of 2½-inch angle-valve connections to the top of the settling tanks.

In order to fill the port side settling tanks a 3½-inch connection is taken from the 4-inch discharge main on the starboard side and run across the forward end of the after boiler room, where it terminates with a 2½-inch connection to each of the port tanks, one of these connections being at the forward end of the after boiler room and the other being at the after end of the forward boiler room, a branch being led through the bulkhead to suit this condition.

Each of the four settling tanks is provided with two 3-inch air-escape pipes, one being at the forward end and the other at the after end of each compartment. These pipes terminate

with automatic valves fitted with wire-mesh screen, located to suit conditions.

Two manholes are also provided in each settling tank. It will be noted that these two low-pressure pumps draw oil from various compartments and discharge the same into the settling tanks. They have a connection for pumping from an oil barge alongside and also a connection (at forward end of aft fireroom) for discharging by means of a hose to a boat or barge alongside. They have no other connections.

It is further to be noted that no cofferdams exist between the settling tanks and the sides of the boilers; in fact, the only cofferdam is between the forward end of the boiler room and the oil tanks at the forward end of the vessel.

DESCRIPTION OF OIL-BURNING SYSTEM.

This system is known as the Koerting System, and was furnished by the Schutte-Koerting Co., of Philadelphia, Pa.

Plate II shows the general arrangement of burners and their connections on the front of the boiler, while Plate III shows a sectional view through the boiler front and gives a clear idea of the principal features, such as air cones, adjustable air registers, and the detachable oil burners, which are of special importance. A photographic view of air register and burner is shown on Plate IV.

The fuel oil is taken, by means of two special supply pumps, from the cargo tanks, and discharged into the settling tanks located at the sides of the boilers. The settling tanks are each provided with a copper steam coil, surrounding the fuel-oil service-pump suction pipes, to increase the temperature of the surrounding oil, and thus render the operation of pumping more effective.

Each boiler room is provided with a separate and complete oil-fuel equipment, in duplicate; consisting of two duplex oil-service pumps, size $4\frac{1}{2} \times 3 \times 4$ inches; two oil heaters of the corrugated film type, each capable of handling 9,000 pounds of oil per hour; one duplex discharge strainer of very fine

mesh; two duplex discharge cocks; two "Macomb" suction strainers, one located at the extreme after end of each settling tank.

The system is so arranged that it is possible to draw the oil from one or the other of the settling tanks through the "Macomb" suction strainers, or from both at the same time, and discharge it by means of one or the other, or both service pumps, through the duplex discharge strainers by means of a by-pass around the heater and then direct to furnaces, or, when working the oil through the heaters, it is possible to use each heater singly or to run them in parallel.

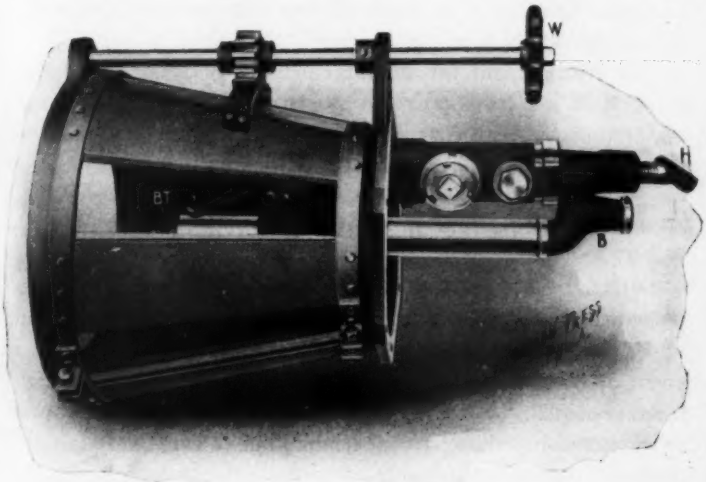


Plate IV.—AIR REGISTER AND BURNER.

Photographic view showing oil burner B, with adjustable air register, provided with air admission slides to properly regulate the necessary air supply for the atomized oil so as to get a perfect mixture of air and oil, and thus secure smokeless combustion.

This arrangement facilitates the cleaning of strainers or of making repairs, should the same be necessary, while the system is operating under normal working conditions. The supply pumps are fitted with pump governors in their steam lines.

There are eleven burners in each boiler, arranged so that they may be readily disconnected or replaced at will; each burner has its individual air register or device for regulating the amount of air to insure a proper mixture for perfect combustion.

The design of these oil burners represents years of experience with spraying apparatus, and seems to be the simplest form possible to construct.

The location of burners is such as will enable the oil to be properly burned before it comes in contact with the cooler surfaces and thus eliminate carbon deposit on tubes, which has been the cause of so much adverse comment on oil burning in boilers of this type.

The arrangement for starting the fires under a cold boiler consists of a triplex pressure hand pump for circulating the oil, and also a small charcoal-fired coil heater for bringing the temperature of the oil to the point of requirement.

INSTRUCTIONS FOR OPERATING.

Lighting Fires.—(A) Use a burner with smallest tip furnished, which is 1½ millimeters, place in center of the boiler on upper row, start hand pump, and when pressure on oil gage on front of boiler shows 100 pounds open burner and light with torch provided. Some little difficulty in igniting may be found when not using the charcoal heater, but it can be done with cold oil.

(B) As soon as sufficient steam is raised in the boiler, start a blower and bring the air pressure up to an equivalent of 3 inches of water pressure, then turn steam on the oil-service pump.

(C) One burner of 1½-millimeter opening being in use, the other burners, which are 2.3 mm., can be cut in, one at a time, always increasing the speed of the blower so as to maintain a proper air pressure, which should be about 4 inches when all burners are in use, and 100 pounds oil pressure is shown on oil gage at the boiler front. Under these conditions air regis-

ters should be closed to about $\frac{3}{8}$-inch mean opening. Oil heaters should be cut in and a temperature of 200 degrees should be used at burners.

(D) If more steam is required the oil pressure should be gradually increased from 100 pounds up to 150 pounds on all burners. This will require the blowers being speeded up in advance of the oil pressure so that when all burners are in use and the oil pressure is about 150 pounds, an air pressure of about 5 inches will suffice. Air registers should be open about $\frac{3}{8}$ inch under these conditions; likewise, two blowers will be needed on account of the loss of air which passes up through the other boiler which is not in use.

(E) If vibration occurs when a burner is being cut in, stop it by immediately cutting out this burner, or two burners can be cut out, until the air pressure can be raised, after which it will cease.

Under no condition should vibration be allowed to continue, as it is due to the air not having sufficient velocity.

(F) An increase in air pressure above 5 inches will be found desirable when working under an oil pressure above 130 pounds, and under no circumstances should the air pressure be restricted, as it can do no harm, and usually proves beneficial. In fact, a high air pressure will enable less air to be used per pound of oil.

(G) When necessary to light up the other boiler in the same compartment, it will be necessary to first increase the speed of the blower, then open up air register to about $\frac{3}{8}$ inch, next turn on oil at burners *one at a time*. In this manner all burners may be cut in and no vibration will be noticed. Don't open up air registers until ready to cut in its own oil burner. Keep watch on the water gage, which records the air pressure, until satisfactory conditions are obtained, after which no trouble will be experienced without change of conditions. As heated oil is being supplied to this boiler, no difficulty will be found in lighting.

(H) With one boiler only in use the other boiler should

be closed up as tightly as possible, so as not to waste the air in the fireroom.

(I) If two boilers are in use in one compartment they should always have the same number of burners in use, and the air registers should be kept in similar positions.

(J) When less than the full number of burners are in use in any boiler the upper wing burners are the most important to cut out, then the lower wing burners are next in importance to leave out; this keeps the oil spray away from the cold tubes and prevents carbon being formed on tubes.

(K) At full power, when it becomes necessary to trim ventilator cowls away from the wind, this should be done only after the blowers have been speeded up enough to increase the air pressure about $\frac{3}{4}$ inch. About 30 revolutions should accomplish this. After the ventilators have been trimmed, it will be found that the pressure has dropped back to about what it was before the blowers were speeded up.

Emergency.—When emergency occurs, requiring the cutting out of a boiler, close master valve, shutting off all burners. If two boilers are in use and only one is to be cut out, the fireman on duty should close each oil burner, at the same time shutting off the air registers. The air pressure should be kept up until all burners in the one boiler are cut out, then the revolutions of the blowers reduced so as to bring down the air pressure to the required amount.

It will be found that burners work very well with air registers open about $\frac{5}{8}$ of an inch and an air pressure of from 4 to 6 inches, depending on oil pressure carried, viz: 4-inch air pressure will do for 100 pounds oil pressure, while 6-inch air pressure will be found desirable when the oil pressure is 175 pounds.

NOTES.

With special reference to oil-fuel equipment on *McCall* and *Burrows*.

On one occasion eleven S. & K. burners were taken out of a boiler while under steam and replaced with eleven others in nine minutes.

Ash pans should have a double set of bricks, laid loose on ½-inch asbestos boards. No sand should be used, as it becomes oil-soaked, and when the oil becomes heated explosions follow, throwing the bricks out of place. All joints in brick work should be free and the top course should overlap the bottom course. Brick work at the sides, back and front of the boiler should be of special tile 4½ inches thick, well secured by bolts in every brick. These tile could be hollowed out in order to reduce weight and they could also have lap-joints to prevent flame passing between joints.

Burners which may be adjusted to suit the best position as regards location from the inside of the boiler are undoubtedly very good for trial as, when once adjusted to their proper length, fixed burners can be furnished according to the length found best by experience.

Large air chambers on oil pumps are a great factor, but air chambers of the dead ends of the pipe line leading across the front of the boiler are of the greatest possible value when used in conjunction with the air chamber at the pump. When these air chambers are properly charged with air all fluctuations in the oil gages disappear. The air chambers on the end of the oil line, having practically the same oil in contact with the contained air at all times, naturally prevent the oil from absorbing as much air as though the oil were passing continuously by the air chamber, as in the case of an air chamber located alongside of a pump. If brass air chambers are used they should be tinned inside to prevent air filtering through.

A high velocity of air undoubtedly works better than a low velocity of air in conjunction with oil fuel, and much less air can be used per pound of oil than in any other manner, as has been shown by test. There should be no limit to the fire-room air pressure on destroyers as it would be impossible for the blowers at present fitted to give a pressure of more than 8 inches or 9 inches of water with everything almost closed. The limit on the blowers should be confined to that of revolutions of the fan, which are about 1,600 per minute maximum. A high velocity of air has another advantage in that should a

boiler tube give away suddenly, the likelihood of scalding those in the fireroom is lessened. Because of liability to accidents of this nature there should always be a blower in operation when underway. In burning oil fuel it is absolutely necessary that blowers be controlled from the fireroom floor; moreover, they should be placed in a convenient position, which adds greatly to safety and to satisfactory operation under all circumstances.

It is claimed that oil with a fire test of 180 to 200 degrees F. is as safe as coal, which will sometimes ignite spontaneously, and at 250 to 300 degrees a red-hot poker will not ignite oil stirred by it, neither will hot coals do so if thrown into it. This, however, does not justify other than the greatest precautions being taken at all times when using or handling fuel oil.

The heating of the air supply is of the greatest possible importance, but where saving in weight is an important factor heating appliances have to be omitted.

In burning oil on destroyers the stack temperature has little effect on the movement of the gases outward, as there is only about 30 feet above the burners. This necessitates forcing all the air necessary for complete combustion between the tubes of the boilers, then through the uptake and up the smokepipe. The greatest resistance being that of passing between the tubes, it therefore becomes necessary to have sufficient fan power to overcome this resistance. It may be argued that a low air pressure is more in line with good engineering than the high air pressure, and ordinarily this is so, as all know that it requires less work on the blower engines to deliver a certain quantity of air at 2 inches of water pressure than it does at 4 inches of water pressure. From experiments, however, it has been found that the amount of air required per pound of oil can be materially reduced by using a high velocity, and this high velocity, with its corresponding volume, has actually been delivered to the boilers with a less total H.P. of the blower engine. If destroyers were intended to be operated at their maximum speed at all times it would naturally be the

endeavor to so proportion smokepipes, uptakes and spaces between boiler tubes so as to burn oil in the most efficient manner at that speed and to do so with the least amount of work on the blower engines; but when it is considered that a torpedo destroyer is actually built to maintain its maximum speed for short periods only, it seems unwise to carry around during the entire life of the vessel the extra weight involved in proportioning uptakes, smokepipes, etc., so as to secure great economy at top speed. In fact the boilers themselves are so designed as to be extravagant at high speed and economical at lower speed; it is perfectly consistent, therefore, that all parts which go to make up an installation of this character should be proportioned with the same object in view. The extra weight required in boilers designed for high efficiency at high speeds is prohibitive in vessels of this class.

DESCRIPTION AND TRIAL DATA OF TORPEDO-BOAT DESTROYERS *PERKINS* AND *STERETT*.

By Captain William Nelson Little, U. S. N.,
Member.

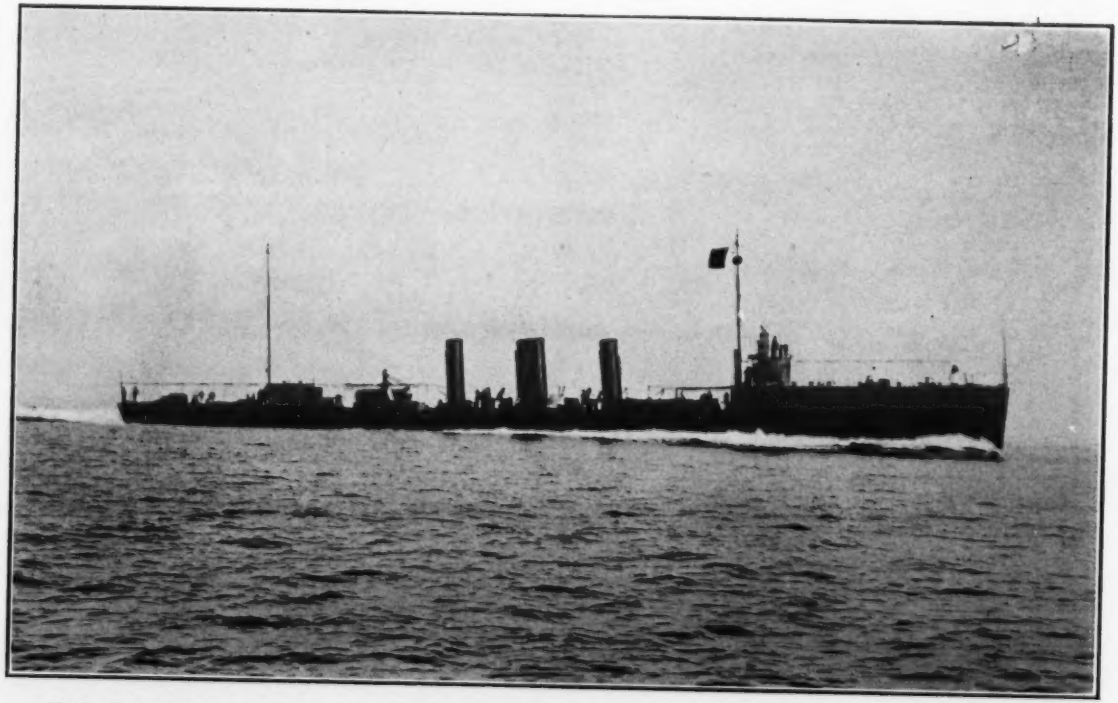
Of the ten destroyers authorized by Congressional Act, approved May 13, 1908, contracts for the construction of two of these vessels were placed with the Fore River Shipbuilding Company and signed October 1, 1908. In general hull characteristics and dimensions the *Perkins* and *Sterett* closely follow the *Flusser* and *Reid*, described in Vol. XXI, No. 1, but differ slightly in draught and displacement and in interior subdivision, due principally to the fact that these latter vessels are fitted solely for oil burning in the generation of steam.

The keels of both vessels were laid on March 22, 1909; the *Perkins* was launched on April 19, and the *Sterett* May 12, 1910, and the vessels turned over to the Government on November 15 and December 12, 1910, respectively. The contract price was $610,000 for each vessel (hull, equipment and installation of armament, $260,000; machinery, $350,000) and the stipulated time of delivery 24 months after date of contract.

Each vessel is of 742 tons' trial displacement, built upon hull-type plans furnished by the Navy Department, with contractors' design of propelling machinery, consisting of two Curtis marine turbines, driving twin screws. Steam is supplied by four water-tube boilers of the Yarrow type, located in two separate compartments. The general arrangement of machinery and principal piping is shown on Plate 3.

CONTRACT TRIALS AND REQUIREMENTS.

In the contract provision was made for certain trials, with guarantees as to steam economy of the machinery installations which had to be satisfactorily fulfilled before Government



U. S. Torpedo-Boat Destroyer "Sterett" on Measured-Mile Course off Rockland, Me., During Official Acceptance Trials.

acceptance of the vessels. The stipulated conditions may be summarized as follows:

(*a*) A progressive speed trial over a measured-mile course for standardizing the screws, extending from maximum speed down to a speed of twelve knots.

(*b*) A full-speed trial of four hours' duration in the open sea, at the highest speed attainable, which must average at least 29½ knots an hour.

During this trial the air pressure in the firerooms is to be limited to an average of 5 inches of water, and the steam pressure at the high-pressure turbine steam chest to 250 per square inch above the atmosphere. The steam consumption of all machinery (including all auxiliaries except evaporating plant), as measured by suitable apparatus temporarily installed for the purpose, must not exceed 14.4 pounds per hour per shaft horsepower developed by the main propelling turbines.

(*c*) An endurance and fuel-oil and steam-consumption trial of twelve hours' duration, in the open sea, at a speed as nearly uniform as possible, and which shall average not less than 25 knots.

On this trial necessary auxiliaries shall be in operation, including those for efficiently lighting the vessel under service cruising conditions, as well as the evaporating and distilling plant, which shall be run at normal-rated capacity. The pumping and forced-draft air systems are also to be operated as required. Under the conditions stated, the steam consumption (excluding steam used in evaporating plant) must not exceed 15.68 pounds per shaft horsepower of the main propelling machinery.

(*d*) A trial of twelve hours' duration in the open sea, at an average uniform speed of 16 knots, as nearly as possible under service cruising conditions.

For this trial the steam consumption must not exceed 21.82 pounds per shaft horsepower of the main propelling turbines.

(*e*) All the above trials to be conducted with the vessel weighted to a mean displacement representing the finished vessel, with equipment, battery, etc., and 90 tons of fuel oil on board, and with two-thirds full supply of stores.

DESCRIPTION OF VESSELS.

Full hull particulars for vessels of this class have heretofore appeared in detail in the JOURNAL. It will only be necessary, therefore, to set forth briefly the important dimensions:

HULL DATA.

Length between perpendiculars, feet	289
over all, feet and inches	293–10½
on load water line, feet	289
Beam, extreme (over guards), feet	27
on load water line, feet and inch	26–00½
molded, feet and inches	26–04½

Draught, normal, forward, feet and inches	8–01½
aft	8–06½
mean	8–04
Displacement, designed, tons	743.5
per inch at load water line at 8 feet 4 inches draught, tons	11.9
Designed draught:	
Area immersed load water line, square feet	144.96
load water line, plane, square feet	5,030
Wetted surface, square feet	7,800
Coefficient of fineness, block	.4142
midship section	.666
L.W.L.	.649

BATTERY.

5 3-inch (14-pounder) semi-automatic rifles, 50 caliber.

TORPEDO TUBES.

3 twin, 5.2-m. × 45-cm., Mark IV, on deck, located as follows:
No. 1, main deck, frame 97, starboard.
No. 2, main deck, frame 97, port.
No. 3, main deck, frame 138, C. L.
2 .30-caliber automatic guns on deck at frame 29, starboard and port.
6 Bliss-Levitt torpedoes, carried in tubes.

COMPLEMENT.

1 commanding officer.
3 ward-room officers.
8 chief petty officers.
16 petty officers.
62 crew.
Total on board, 90.

MACHINERY.

MAIN TURBINES.

The *Perkins* and *Sterett* are propelled by twin screws driven by 72-inch Curtis marine turbines, one on each shaft, having 14 ahead and 2 reverse stages. Plate 1 shows a view of one of the starboard turbines assembled complete, with top casing removed. The steam opening to each main turbine is 9 inches in diameter; area of exhaust opening, 16.05 square feet.

Each turbine consists essentially of the stator (or cylinder) and the rotor (or wheels and drum). The cylinders are of

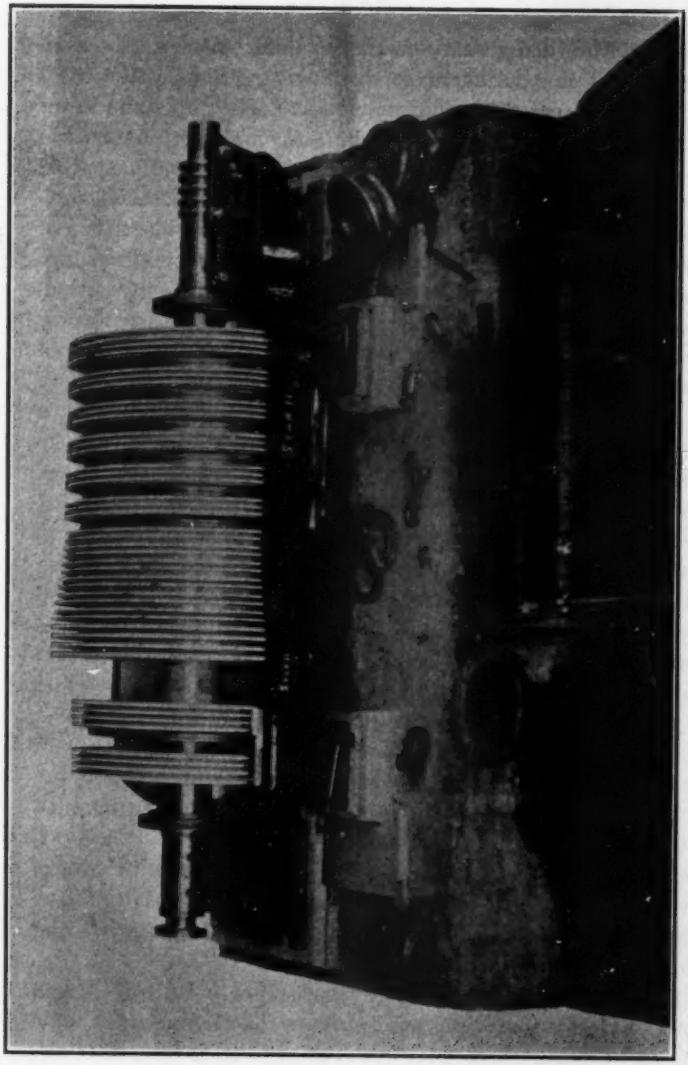


Plate I.—Photographic View of Starboard Main Turbine with Top Casing Removed. U. S. Torpedo-Boat Destroyers "Perkins" and "Sterett."

cast iron, smoothly bored inside. They are cast in four sections and bolted together. The brackets for the rotor bearings and the thrust bearings are separate castings, bolted to the ends of the cylinders, at the lower sections, which construction differs from previous practice. The upper and lower sections part on a horizontal plane through the center line of rotor shaft. Annular circumferential ribs equally spaced, grooved on their inner peripheries to take the beaded peripheries of diaphragms, are cast with the forward sections of each cylinder. Diaphragms and ribs divide the cylinders into sections or stages, of which in the forward half casing or cylinder there are six. These ribs also support the nozzles for each stage, and are provided with openings for the passage of steam to the nozzles.

Each cylinder is provided with an ahead and an astern steam chest, in which are placed nozzle ports and a valve to each ahead nozzle. In the ahead chest there are eighteen, and in the astern chest nine, nozzles. The ahead valves are operated individually by stems extending through the casing. Two nozzles only in each astern chest are fitted with closing valves.

Each successive stage has an increased number of nozzles symmetrically placed each side of the vertical axis, such that on leaving the sixth stage steam passes through nozzles placed all the way round the diaphragm.

The rotor consists of six ahead and two astern wheels and a drum, all mounted on one shaft and in one casing.

The diaphragms are built up of Class B steel plate, rimmed with forged-steel rings and turned to fit grooves in the cylinder partitions. They are bored and bushed with bronze sleeves of such construction that a slight eccentricity of the rotor shaft will be accommodated by corresponding yielding of the rubbing part of the bushing.

Stage Wheel.—Each wheel consists of a hub of forged steel, which is forced on and keyed to the turbine shaft, having eight radial arms or stiffeners of mild steel riveted to the

hub; a rim of medium forged steel encircling the arms; two parallel circular side plates or discs of rolled steel, riveted and otherwise secured to hubs and to rims; bucket holders of cold-rolled steel, inverted rectangular U-section in segments calked into the rim; buckets of composition (extruded metal) riveted into the aforesaid segments, and outside of all, the shrouds or bands of sheet steel, into which are riveted spurs of the tips of the buckets.

The bucket holders are milled from solid steel bars of rectangular section. After milling the exterior faces of the bars a rectangular groove is milled, thus forming a channel bar with a thin web, which latter is slotted to receive, and into which are riveted, spurs of the buckets. After the blades are riveted into their holders and segments the latter are sawed to permit of bending to the curvature of the wheel rim. Circumferential grooves or channels in the rim receive the corresponding parts of the segments. Shallow longitudinal grooves are cut during the process of machining, one on each side of the holder, into which the metal of the rim is calked.

Reversing wheels, of 55-inch pitch diameter, consist of two stages of four rows of buckets each, which are fixed to the shaft one foot abaft the ahead rotor and within the exhaust space. Their construction is similar to that of the ahead wheels.

The Stator.—The stationary guide blades are made and assembled in a way described for the rotor blading, excepting that segment holders of composition are provided, which are fitted into grooves of the casing and secured there by tap bolts passing through casing from the outside.

The Nozzles.—The first-stage ahead nozzles are rectangular in section, with two diverging walls in each nozzle, such as to make the expanding ratio of outlet to throat section 1.145.

On the post-trial examination of the machinery the first-stage nozzles of the starboard turbine of the *Perkins* were found broken in part on their thin exit edges. The contractors, therefore, substituted Monel metal for bronze for the

casting of manifold of these nozzles, and in later designs have supplied cruising nozzles (two) occupying the same space as the three first original nozzles, and with a throat area of 0.8 square inch each, and an expansion ratio of 2.5 in a length of 1.286 inches.

The expansion nozzles occur only at the first stages ahead and astern; the others, or intermediate nozzles, act as ports and directors, with increased aggregate area of opening. Up to and including the seventh stage, the nozzle plates are of cast iron, with cold-rolled nickel-steel blades as directors. From the seventh stage to the last expansion the nozzle directors are made in segments of bronze and the blades of Monel metal, the latter being molded into the casting.

Important data of each turbine are given below:

Stage.	Buckets.			Intermediates.	Nozzles.		
	Number of rows on wheels.	Width, inches.	Length, inches.	Number of rows.	Number of openings.	Total area of openings, sq. inches.	Arc of nozzles, degrees.
1st	4	1 to .75	1.33 to 3.00	3	19	8.89	42.9
2d	3	.75	1.54 to 2.70	2	33	15.15	66
3d	3	.75	1.54 to 2.70	2	45	20.66	90
4th	3	.75	1.54 to 2.70	2	65	29.84	130
5th	3	.75	1.54 to 2.70	2	95	43.61	190
6th	3	.75	1.54 to 2.70	2	140	64.27	280
7th	2	.75	1.85 to 2.38	1	174	101.6	348
8th	2	.75	2.29 to 2.94	1	136	126.5	360
9th	2	.75	2.99 to 3.84	1	136	165.3	360
10th	2	.75	3.82 to 4.92	1	136	211.7	360
11th	2	.75	4.99 to 6.42	1	136	276.3	360
12th	2	.75	6.53 to 8.40	1	136	361.5	360
13th	2	1	7.93 to 8.92	1	136	479.7	360
14th	2	1	8.09 to 9.10	1	136	619.8	360
1st reverse	4	1 to .75	1.80 to 3.375	3	9	6.075	49
2d reverse	4	1 to .75	2.36 to 4.50	3	58	58.0	336

SHAFTING, BEARINGS, ETC.

	Material, Class.	Length, feet and inches.	Diameter, inches.	Diam. of hole, inches.
Rotor shaft	A steel.	19–08	17, 12, $8\frac{1}{2}$	$10\frac{1}{2}$, $8\frac{1}{2}$, 4
Thrust shaft on rotor shaft.	A steel.	...	$8\frac{1}{2}$	4
Forward line shaft	A steel.	S. 21–06$\frac{1}{2}$ P. 8–04$\frac{7}{16}$	$8\frac{1}{2}$	4
After line shaft	A steel.	S. & P. 12–08$\frac{1}{16}$	$8\frac{1}{2}$	4
Stern-tube shaft	A steel.	S. & P. 23–05$\frac{3}{16}$	$8\frac{1}{2}$, $7\frac{1}{2}$	4, 3, 2 & $1\frac{1}{16}$
Propeller shaft	A steel.	S. & P. 31–03$\frac{1}{2}$	$8\frac{1}{2}$	4, 2

Rotor shaft bearings, number of for each main turbine 2
length of, forward, inches ..... 12 × 21
after, inches ..... 12 × 21

Caps for each bearing are 21 inches long, with 15-inch bearing surface in their length.

Thrust shafts overhung from forward main bearing. No steady bearings provided.

Collars on thrust shaft:

Number	5
Thickness, inches	$1\frac{1}{16}$
Space between, inches	$2\frac{1}{2}$
Outside diameter, inches	$14\frac{1}{2}$
Inside diameter, inches	$8\frac{1}{4}$
Bearing surface, each collar, square inches	107
Thrust shoes, number	4
effective (thrust) surface, square inches	297

Line-shaft bearings:

Number of bearings, starboard	3
port	2
Length of bearings, inches	13

Stern-tube bearings:

Length of forward bearing, inches	24
after bearing, inches	36

Propeller-shaft bearings:

Length of forward strut bearing, inches	28
after strut bearing, inches	34

CONDENSERS AND FEED HEATER.

There are two main condensers, one for each propelling turbine, installed in the engine rooms, and an auxiliary condenser and one feed heater in the starboard engine room, with tube data as follows:

.....	Tubes.				
	No.	Thickness, B.W.G.	Diameter (outside), inch.	Length, feet and inches.	Cooling or heating surface, square feet.
Each main condenser...	2,084	No. 18	$\frac{5}{8}$	14-02	4,773
Auxiliary condenser.....	255	No. 18	$\frac{5}{8}$	6-03$\frac{1}{2}$	256
Feed heater.....	437	No. 18	$\frac{5}{8}$	7-03$\frac{1}{2}$	500

PROPELLER DATA.

Number, type and material: two, 3-bladed, true screw, Monel metal, cast solid.

	Perkins.	*Sterett.*
Diameter, feet and inches.....	6-07	6-06$\frac{1}{4}$
Area, projected, square feet.....	19.22	18.21
helicoidal, square feet.....	22.12	21.06
disc, square feet.....	34.04	33.82
Pitch as measured, mean, inches.....	76.34	75.61
Height of lower tip of blade below keel, inches.....	3	2.88
Immersion of upper tip of blade, at load draught, inches,	21.25	21.38

BOILER DATA.

Number and type: four (two in each boiler compartment), Yarrow, water-tube, arranged for oil burning.

Designed working pressure, pounds.....	265
Diameter (inside) of steam drum, inches.....	43
Water drums, semicircular, with riveted oval tube sheets, major and minor diameters, inches.....	16 × 22
Height above ash pan, feet and inches.....	12-10
Length, external, feet and inches.....	12-09
Width, external, feet and inches.....	14-02$\frac{1}{2}$
Heating surface, each boiler, square feet.....	4,500
Number of burners, each boiler.....	7

Tubes, steel, solid, cold-drawn, seamless, each boiler, as follows:

Number.	Thickness, B.W.G.	Outside diameter, inches.	Length as fitted, feet and inches.
110	11	1$\frac{1}{2}$	7-07$\frac{1}{4}$
110	11	1$\frac{1}{2}$	7-08$\frac{1}{4}$
660	12	1	7-05
495	12	1	7-05$\frac{1}{2}$
330	12	1	7-06
330	12	1	7-06$\frac{1}{2}$
165	12	1	7-07$\frac{1}{4}$
165	12	1	7-08$\frac{1}{4}$

Kind of forced draft: closed firerooms.
Height of smokepipes above ash pans, feet..... 35
Number of smokepipes..... 3
Area of forward smokepipe, square feet..... 13.41
middle* smokepipe, square feet..... 26.52
after smokepipe, square feet..... 13.41

PUMPS AND AUXILIARY ENGINES (EACH VESSEL).

Name.	No.	Type, make and location.	Steam cylinders.				Water cylinders.			
			No.	Diam., in.	Diam. piston rod, in.	Stroke, in.	No.	Diam., in.	Diam. piston rod, in.	Stroke, in.
Pumps:										
Main air.....	2	Vertical, twin, Fore River. Engine room.	2	10	1¼	16	2	22	2½	16
Main circ.....	2	Vertical, direct connected. Engine room.	1	8	1½	8	Centrifugal 28-inch runner.			
Main feed.....	2	Blake, vertical, simplex. Engine room.	1	13	2½	16	1	9	2½	16
Aux. feed	2	Blake, vertical, simplex. One in each fireroom.	1	13	2½	16	1	9	2½	16
Bilge.....	1	Fore River, vertical, simplex. Forward. Compartment A15.	1	3½	⅞	4	1	4	⅞	4
Fire and bilge.	1	Fore River, vertical, simplex. Engine room.	1	10	1½	8	1	7	1½	8
Fire and bilge.	2	Fore River, vertical, simplex. One in each fireroom.	1	7	1½	12	1	5	1½	12
Aux. Cond.....	1	Fore River, horizontal, combined air and circulator. Engine room.	1	6	1½	7	1	8	1½	7
							1	8	1½	7
Oil cooler.....	1	Fore River, vertical, simplex. Engine room.	1	4	⅞	6	1	5	⅞	6
Evap. and Distill. Plant.†	1	Fore River, vert. sim. evap. feed.	1	3½	⅞	4	1	4	⅞	4
		Fore River, vert. sim., fresh water. Engine room.	1	3½	⅞	4	1	4	⅞	4
Fuel-oil services.	4	Fore River, vertical, duplex. Two in each fireroom.	2	4½	⅞	4	2	3	⅞	4
Fuel-oil tank supply.	2	Fore River, vertical, simplex. One in each fireroom.	1	6	1½	12	1	7	1½	12
Lubricating oil.	2	Fore River, vertical, simplex. Engine room.	1	4	⅞	6	1	5	⅞	6
Dynamo engines	2	Horizontal, one-stage Curtis turbine. 5 kw., direct connected. Engine room.								
Forced-draft blowers.	4	12-inch 7-stage Kerr steam turbine and 27-inch Sirocco fan. Two in each fireroom.								

† Maximum rated output of fresh water (distilled) per 24 hours: Evaporators, 3,000 gallons; distillers, 2,000 gallons.

FUEL-OIL SYSTEM.

This system is composed of three storage tanks forward and five aft, four settling tanks, two to each boiler compartment, and the following machinery:

* The middle smokepipe, to which are connected boilers B and C, has a central division plate.

2 fuel-oil supply pumps (2 to a compartment).
4 fuel-oil service pumps to burners.
4 fuel-oil heaters (2 to each boiler compartment).
2 hand-service pumps (1 to a compartment).
28 burners (7 to each boiler).

FUEL-OIL HEATERS.

There are two corrugated spiral-film oil heaters, of the Schutte-Koerting pattern, located in each boiler compartment, and fitted with interlocking plug cocks, so arranged that the heaters may be used either singly or in parallel. The shell of each heater contains four pairs of double-corrugated spiral tubes, secured into composition headers, which latter are so constructed as to admit steam through the interior of the central tubes and outside of the outer tubes.

TRIALS.

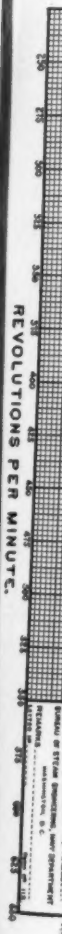
STANDARDIZATION.

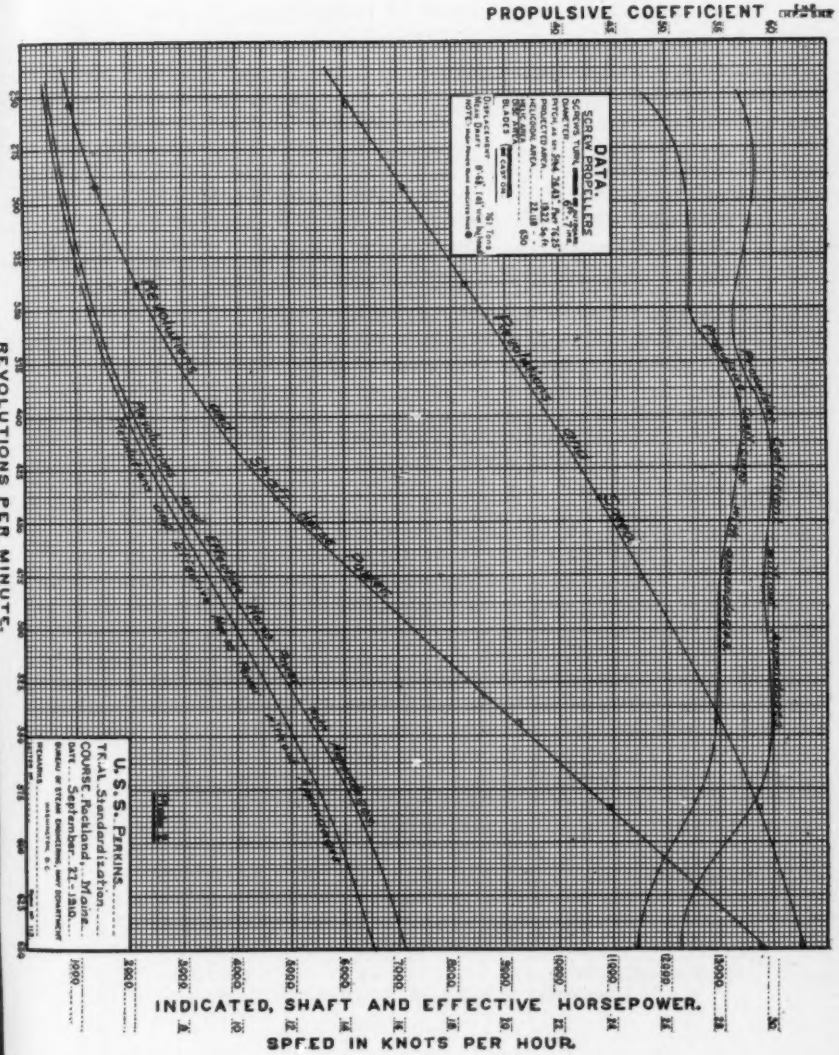
Both vessels were standardized over the Rockland measured-mile course, the *Perkins* on September 27, 1910, and the *Sterett* on November 1, 1910.

Fottinger torsionmeters were used to ascertain power developed on the main shafts. The readings of this instrument on the port shaft of each vessel, however, showed inconsistent and variable results, for which reason they were discarded as unreliable. On the various trials the power developed by the port main turbine was, in consequence, figured from that obtained on the starboard shaft, it being assumed that for either turbine the power was the same when running at the same number of revolutions.

The following is from standardization trials:

	Perkins.	*Sterett.*
Mean draught, feet and inches	8–06½	8–06
Mean displacement, tons	761.0	756.0
Mean revolutions (both shafts) for speed of 16 knots	289.5	293.2
25 knots	475.1	484.5
29.5 knots	584.8	598.8
Speed, average of highest runs, knots	31.01	31.63




PROPULSIVE COEFFICIENT
REVOLUTIONS PER MINUTE.
INDICATED, SHAFT AND EFFECTIVE HORSEPOWER.
SPEED IN KNOTS PER HOUR.
DATA.
SCREW PROPELLERS
U.S.S. PERKINS
TRIAL Standardization
COURSE Rockland, Maine
DATE September 22, 1910

Data of standardization runs are given in Tables A and B, and from the former, curves of performance (*Perkins*) have been laid down as shown on Plate 2.

FOUR-HOUR FULL-SPEED TRIAL.

The *Perkins* ran this trial on September 30, 1910, from 9:10 A. M. to 1:10 P. M., the weather being clear, cool and pleasant, with gentle breeze from the S.W., and smooth sea. The course was along the coast of Maine, New Hampshire and Massachusetts. During the trial auxiliaries were run as follows: (*a*) auxiliaries in connection with the operation of main turbines; (*b*) dynamos as required to light the ship; (*c*) the flushing system; (*d*) the steering enging as required; (*e*) the fuel-oil pumps; (*f*) forced-draft blowers.

The four-hours' full-speed trial of the *Sterett* was run on November 6, 1910, from 7:15 A. M. to 11:15 A. M., the weather was fair and pleasant with moderate to fresh breeze from W.S.W., and a short, choppy sea from the same direction. Auxiliaries were in operation as follows: 2 main feed pumps; 2 main circulating pumps; 4 forced-draft blowers; 1 auxiliary feed pump (standing by); 2 forced-lubrication pumps; 1 fire and bilge pump (sanitary); 2 fuel-oil pumps; 1 fuel-tank pump (as needed); 1 dynamo; 1 steering engine.

The draught was taken at the beginning and end of trials as follows:

	Perkins.	*Sterett.*
Draught, at beginning, forward, feet and inches.....	8–06$\frac{7}{8}$	8–06$\frac{9}{16}$
aft, feet and inches.....	8–09$\frac{1}{2}$	8–07$\frac{15}{16}$
mean, feet and inches.....	8–08$\frac{11}{16}$	8–07$\frac{1}{4}$
Displacement, tons	785.5	774.5
Draught, at end, forward, feet and inches.....	7–11$\frac{7}{8}$	7–11$\frac{7}{16}$
aft, feet and inches.....	8–10$\frac{1}{16}$	8–07$\frac{15}{16}$
mean, feet and inches.....	8–05	8–03$\frac{11}{16}$
Displacement, tons.....	747.0	732.5
estimated, at middle of trial, tons.....	765.0	754.0

Table A.—TURBINE DATA.—STANDARDIZATION TRIALS, U. S. S. "PERKINS," OFF ROCKLAND, ME., SEPT. 27, 1910.

No. of run.	Speed.	Revolutions per minute.			Steam pressures.								Pressure in oiling system (gage).	Vacuum in inches of mercury.		Temperatures.							Torsion meter readings.	Brake horsepower.		
					Engine room, gage.			Ahead, chest (gage.)		First stage ahead (gage).						In engine room.	Injection.	Discharge.		Air-pump discharge.		Feed.	Outside air.				
		Shaft No. 1.	Shaft No. 2.									Nozzles open, each engine.	Temp., main steam.											K = .0734. Shaft No. 1.	Shaft No. 1.	† Shaft No. 2.	
		S.	P.	Average.	Main steam.	Aux. steam.	Aux. exhaust.	P.	S.	P.	S.			P.	S.			P.	S.	P.	S.				S.	P.	Total.
1	12.951	245.72	237.14	241.43	240	250	10	202	202	0	0	1	280	15	28	28	90	54		63	59	59	68	59	22.8	411	365	776
2	15.196	265.94	259.03	262.48	255	252	8	190	195	0	0	1	286	15	28	28.5	86	54		66	59	59	68	59	25.17	491	470	961
3	13.104	252.30	246.71	249.60	240	247	8	175	180	0	0	1	278	15	26	28.5	86	54		65	58	58	68	57	23.45	434	405	839
4	16.898	295.40	290.05	292.72	270	248	9	150	150	.1	0	2	272	15	28	28.5	92	54		67	59	59	68	57	32.14	697	655	1,352
5	15.618	296.89	294 75	295 82	280	258	9	155	150	.1	0	2	276	15	28.4	28.2	92	54		68	59	58	70	57	32.21	702	685	1,387
6	17.045	297.48	293.62	295.80	265	252	8	150	155	.1	0	2	278	15	28.4	28 5	94	54		68	58	58	68	58	31.68	693	675	1.368
7	17.592	335 91	333.52	334.71	265	253	8	130	125	0	0	4	280	15	28 4	28 2	94	54		70	58	58	68	58	41.38	1,021	995	2,016
8	19.452	343.29	337.15	340.22	260	256	6	150	150	0	0	4	280	15	28.2	28.2	94	54		71	60	58	68	57	42.64	1,075	1,030	2,105
9	17.887	341 65	336 64	339.14	260	255	7	155	155	0	0	4	278	15	28.3	28.2	94	54		71	58	58	70	57	42.50	1,067	1,025	2,092
10	24.000	436.0	434.62	435.31	260	255	4	245	245	22	22	5	410	15	28.2	28.2	92	54		77	62	60	72	57	73.09	2,342	2,305	4,647
11	22 794	438.18	438 08	438.13	266	262	5	245	245	22	22	5	410	15	28 4	28.5	90	54		76	62	59	74	57	73 45	2,364	2,365	4,729
12	24.110	439 61	437.20	438.40	265	260	5	240	245	22	22	5	410	15	28.5	28.0	96	55		76	60	62	74	57	74.59	2,408	2,350	4,758
13	24.294	478.01	475.88	476.95	265	260	6	220	220	33	33	7	408	15	28.0	28.2	96	55		79	60	64	75	57	90 28	3,169	3,110	6,279
14	25.756	476.36	474.25	475 30	265	260	5	225	223	33	33	7	408	15	28.2	28.4	98	55		80	63	64	74	58	89.56	3,133	3,075	6,208
15	24.379	479.30	475 95	477 63	251	252	5	220	220	32	35	7	408	15	28 2	28.0	98	55		80	60	64	74	59	90.40	3,182	3,120	6,302
16	28.835	546 06	546.11	546 08	255	248	10	245	245	52	52	9	412	15	28.2	28.2	100	55		84	65	70	74	60	115 8	4,643	4,695	9,338
17	27 136	540 97	542 24	541 61	265	260	6	250	250	54	55	9	410	15	28.2	28 0	100	55		83	70	70	72	59	115.9	4,604	4,600	9,204
18	28.720	542.88	543.00	542.94	260	255	10	245	245	53	54	9	410	15	28.0	28.0	102	55		83	70	65	74	59	116.0	4,624	4,630	9,254
19	28.697	574.42	575.12	574.77	245	243	9	225	225	62	62	11	406	15	28.0	28 0	102	55		83	70	70	76	59	124.9	5,269	5,300	10,569
20	29.967	583 74	583 84	583 79	250	235	8	245	245	65	65	11	410	15	28.0	28.0	104	55		83	70	70	126	59	228.04	5,491	5,485	10,976
21	29.173	589 74	590.28	590.01	255	250	9	245	245	67	67	11	410	15	28.0	28.0	108	55		82	70	68	124	59	129.8	5,623	5,625	11,248
22	31.383	637 16	639.04	638.10	220	215	8	215	215	85	85	15	400	15	28.0	28.0	110	55		86	70	72	118	60	142.04	6,648	6,660	13,308
23	30 054	636.63	640.76	638.69	235	235	6	225	225	90	90	15	400	15	28.0	28.2	110	55		86	70	68	118	60	143.85	6,726	6,695	13,421
24	31.854	664 90	669.24	665 07	240	245	9	235	235	94	94	15	408	15	28.0	28.2	108	55		86	70	72	120	59	146.58	7,156	7,325	14,481
25	30.656	645.82	645.72	645.77	240	238	8	230	230	90	90	15	400	15	28.0	28.0	108	55		86	70	70	118	57	143.14	6,791	6,800	13,591
26	31.641	660.59	662.19	661.39	250	245	7	240	240	95	95	15	408	15	28.0	28.0	108	55		86	70	70	120	57	146 21	7,092	7,170	14,262

* Speed uncorrected for tide.

† Shaft No. 2, calculated.

Table B.—Turbine Data.—Standardization Trials, U. S. S. "Sterett," off Rockland, Maine, Nov. 1, 1910.

No. of run.	Speed.*	Revolutions per minute.			Steam pressures.								Pressure in oiling system, gage.	Vacuum in inches of mercury.		Temperatures.								Torsion meter readings.	Brake horsepower.		
					Engine room, gage.											In engine room.	Injection.	Discharge.		Air-pump discharge.		Feed.	Outside air.				
		Shaft No. 1.	Shaft No. 2.	Average.	Starboard (main steam).	Port (main steam).	Auxiliary steam.	Auxiliary exhaust.	Ahead chest, gage.		First stage ahead, gage.		Nozzles open, each engine.											K = .0732. Shaft No. 1.	Shaft No. 1.	†Shaft No. 2.	Total.
		S.	P.						P.	S.	P.	S.		P.	S.			P.	S.	P.	S.			S.	S.	P.	
1	11.840	209.33	205.31	207.32	248	262	253	5	138	141	1	0	1	14	29	29	64	50	54	55	52	51	83	46	16.61	255	240	495
2	11.048	209.70	206.72	208.21	251	265	260	4.2	140	136	1	0	1	14	29	28.7	64	50	55	54	52	50	84	46	16.59	255	250	505
3	11 997	213.18	207.51	210 35	250	261	257	4	143	138	1	0	1	14	29	29	64	50	55	55	52	50	83	50	17.00	265	255	520
4	14.996	285.87	275.69	280.78	256	266	258	7.5	180	175	3	5	1	14	28 7	29	64	50	58	58	52	50	84	44	29.87	625	575	1,200
5	15.875	290.20	278 61	284.40	260	273	265	7	187	178	5	5	1	14	28.6	29	65	50	58	60	52	52	84	48	31.08	660	590	1,250
6	15.555	295.83	286.34	291.49	254	260	255	8.5	190	190	5	6	1	14	28.5	28.9	66	50	58	61	53	52	84	46	32 54	705	635	1,340
7	20.· 77	368.11	363.52	365.81	241	250	250	5	200	194	15	10	4	15	28.7	28.8	67	50	73	70	53	53	84	48	48.25	1,300	1,240	2,540
8	20.066	378.16	378.01	378 08	245	255	250	5	205	200	15	10	4	15	28.8	28 8	67	50	70	70	53	53	68	48	50.42	1,396	1,400	2,796
9	20.607	379.75	376.65	378.20	255	270	260	4	205	205	13	11	4	15	28 8	28 8	66	50	69	69	54	53	84	48	51.50	1,431	1,385	2,816
10	22.747	437.03	434.77	435 90	250	255	250	6	245	250	25	22	5	15	28.5	28.5	67	50	75	76	56	54	91	48	69.86	2,235	2,195	4,430
11	22.898	434.75	431.85	433.30	240	250	245	6.5	242	240	25	22	5	15	28.5	28.5	67	50	75	76	56	56	93	48	68.95	2,194	2,150	4,344
12	23.028	440 67	437.72	439.20	250	270	260	7.5	250	245	26	24	5	15	28.5	28.7	66	50	74	76	58	58	60	48	71.05	2,292	2,240	4,532
13	24.852	482.08	477 90	479.99	230	245	235	7	235	225	40	35	7	15	28.5	28.4	65	50	75	79	60	60	62	48	86.45	3,050	2,960	6,010
14	25.221	491.54	485.14	488.34	255	275	265	5.5	246	230	40	35	7	14	28 8	28.8	66	50	78	78	63	63	122	48	90.16	3,243	3,100	6,343
15	25.101	493 02	485.37	489.20	250	265	255	5.5	240	235	40	37	7	13	28.5	28 5	64	50	78	78	63	63	120	48	90.95	3,282	3,110	6,392
16	25 237	500.28	493.99	493 99	250	270	260	5 5	250	245	43	38	7	15	28.7	28.8	64	50	79	79	63	63	119	50	94·3	3,453	3,295	6,748
17	25.825	498.60	491.86	495.23	250	265	255	5	250	245	43	37	7	15	28.7	28.8	66	50	78	78	63	63	144	50	93·3	3,405	3,245	6,650
18	25.112	500.21	493.42	496.71	258	270	265	5.5	250	250	43	38	7	15	28 7	28.8	66	50	78	78	63	63	142	50	93·7	3,430	3,275	6,705
19	...	...	...	...	255	268	265	6.5	245	245	58	50	9	15	28.5	28.5	65	50	82	80	65	65	140	50	110.0	...	...	...
20	...	...	...	...	245	258	255	5.5	245	240	56	50	9	15	28.5	28.4	78	50	82	81	65	65	144	50	109.4	...	...	...
21	...	...	...	...	245	255	250	6	243	240	58	50	9	15	28 4	28.3	82	50	82	78	70	70	134	50	109.2	...	...	...
22	28 335	553.40	535.63	544.52	245	258	255	6	245	240	60	53	9	15	28.4	28.4	82	50	82	82	70	70	130	52	116.95	4,736	4,320	9,056
23	27.302	559.82	541.99	550.90	250	260	260	7	250	245	58	57	9	15	28 4	28.2	82	50	82	81	70	70	136	50	118.70	4,863	4.475	9,338
24	28 601	562 24	543.58	552.91	260	275	265	7	250	247	68	58	9	15	28.5	28.2	82	50	82	82	70	70	141	50	120.0	4,939	4,510	9.449
25	28 173	585.91	567 34	576.62	235	250	245	10	220	210	69	68	12	15	28.4	28.2	82	51	83	83	70	70	136	50	125.1	5,364	5,015	10,379
26	29.617	591.73	573.16	582.45	236	250	245	11.5	220	210	69	68	12	15	28.2	28.2	79	51	83	83	70	70	142	52	127.04	5,502	5,130	10,632
27	28.513	593.66	575.47	584.57	248	255	250	12.5	230	230	70	68	12	15	28.2	28.2	80	51	82	83	70	70	142	52	127.0	5,517	5,175	10,692
28	...	644.22	612.66	628.44	210	235	245	11	210	200	80	65	14	15	28	28	78	51	87	85	73	73	139	52	137.4	6,479	...	...
29	...	658 10	621.38	639.75	235	250	248	12.5	225	225	88	85	14	15	28	28	81	51	88	86	80	80	142	52	138 5	6,671	...	...
30	30.034	701.35	666.28	683.81	260	260	260	17	240	250	102	101	15	15	27.9	27 5	78	51	90	88	82	82	160	52	148.95	7,646	6,915	14,561
31	31.236	699 04	663.07	681.05	250	258	255	16	230	240	100	98	15	14	27.9	27.6	81	51	90	88	82	82	144	52	146.3	7.485	6,850	14,335
32	31.816	693.37	658.39	675.88	248	250	250	15.5	230	240	100	98	14	15	27.9	27.7	78	51	89	88	80	80	146	52	147.32	7.475	6,765	14,240
33	31.295	7.4.58	672.33	688.46	255	250	260	16	235	245	101	99	15	15	27.9	27.9	78	51	90	88	80	80	148	50	149.20	7 695	7,030	14.725
34	32.333	707.50	683.38	695.44	250	260	260	16	250	245	108	100	15	15	27 8	27.8	78	51	90	88	82	81	148	50	149.0	7,715	7,245	14,960

*Speed uncorrected for tide.

†Shaft No. 2 calculated.

PERFORMANCE FOUR-HOUR FULL-SPEED TRIALS.

	Perkins.	*Sterett.*
Average steam pressures :		
At boiler, gage, pounds	257.0	263.0
Engine room, main steam, gage, pounds	238.0	250.0
auxiliary steam, gage, pounds	235.0	251.0
Auxiliary exhaust, gage, pounds	6.2	13.6
Starboard steam chest, gage, pounds	231.0	231.0
Port steam chest, gage, pounds	231.0	226.0
Starboard, 1st stage, gage, pounds	73.7	86.5
Port, 1st stage, gage, pounds	70.6	87.8
Number of nozzles open on H.P. steam chest	12.2	14.0
Vacuum in condenser, starboard, inches of mercury	28.0	26.8
Vacuum in condenser, port, inches of mercury	27.6	27.7
Air pressure in fireroom, forward, inches of water	2.8	3.3
aft, inches of water	2.1	3.3
Barometer	30.1	29.8
Revolutions, or double strokes, per minute :		
Shaft No. 1, starboard	595.5	644.5
No. 2, port	591.5	617.8
Average	593.5	631.2
Pumps, main-air, starboard	29.8	32.5
port	29.0	33.1
circulating, starboard	174.0	180.7
port	185.6	133.0
feed, forward	26.2	25.1
aft	26.8	27.8
forced-lubrication	17.0	24.0
fuel-oil, forward	41.0	46.4
aft	37.5	47.4
Forced-draft blowers (average of four)	1,413.0	1,537.0
Speed of ship, knots	29.76	30.37
Slip of propeller, in per cent. of its own speed, based on mean pitch :		
Shaft No. 1, starboard	20.5	24.3
No. 2, port	19.7	20.9
Mean	20.1	22.6
Shaft horsepower :		
(*a*) Total	11,668.0	12,789.0
(*b*) From standardization curve at revolutions corresponding to trial speed	11,550.0	13,000.0
Temperatures, degrees Fahrenheit :		
Injection	57.4	50.0
Discharge	86.2	88.8
Air pumps, discharge	74.0	80.0
*Feed	...	...

* Feed heater not in use : auxiliary exhaust directed into main turbines.

† Shaft No. 2 calculated.

• Speed uncorrected for tide.

	Perkins.	*Sterett.*
Temperatures, degrees Fahrenheit:		
Outside air	59.5	47.1
Engine room, working level	92.5	85.9
Fireroom, working level	82.1	77.9
Fuel oil:		
Specific gravity	0.874	0.884
Burners in use, each boiler	7.0	7.0
Pressure at burners, gage	99.8	101.0
Temperature at burners, degrees F	147.2	...
(*m*) Pounds per hour	16,400.0	18,223.0
Knots per ton (2,240)	4.065	3.733
Water:		
(*g*) Pounds per hour, all machinery	169,083.0	190,017.0
per S.H.P. main engine	14.49	14.85

DEDUCED DATA.

(*q*) Square feet of heating surface per S.H.P.	1.54	1.407
(*r*) Pounds of oil per S.H.P. per hour	1.40	1.428
(*u*) Cooling surface (main condensers), square feet per S.H.P.	.818	.746
(*y*) Heating surface used during trial, square feet	18,000.0	18,000.0
(*z*) Pounds of water evaporated per pound of oil $\frac{(g)}{(m)}$	10.31	10.42

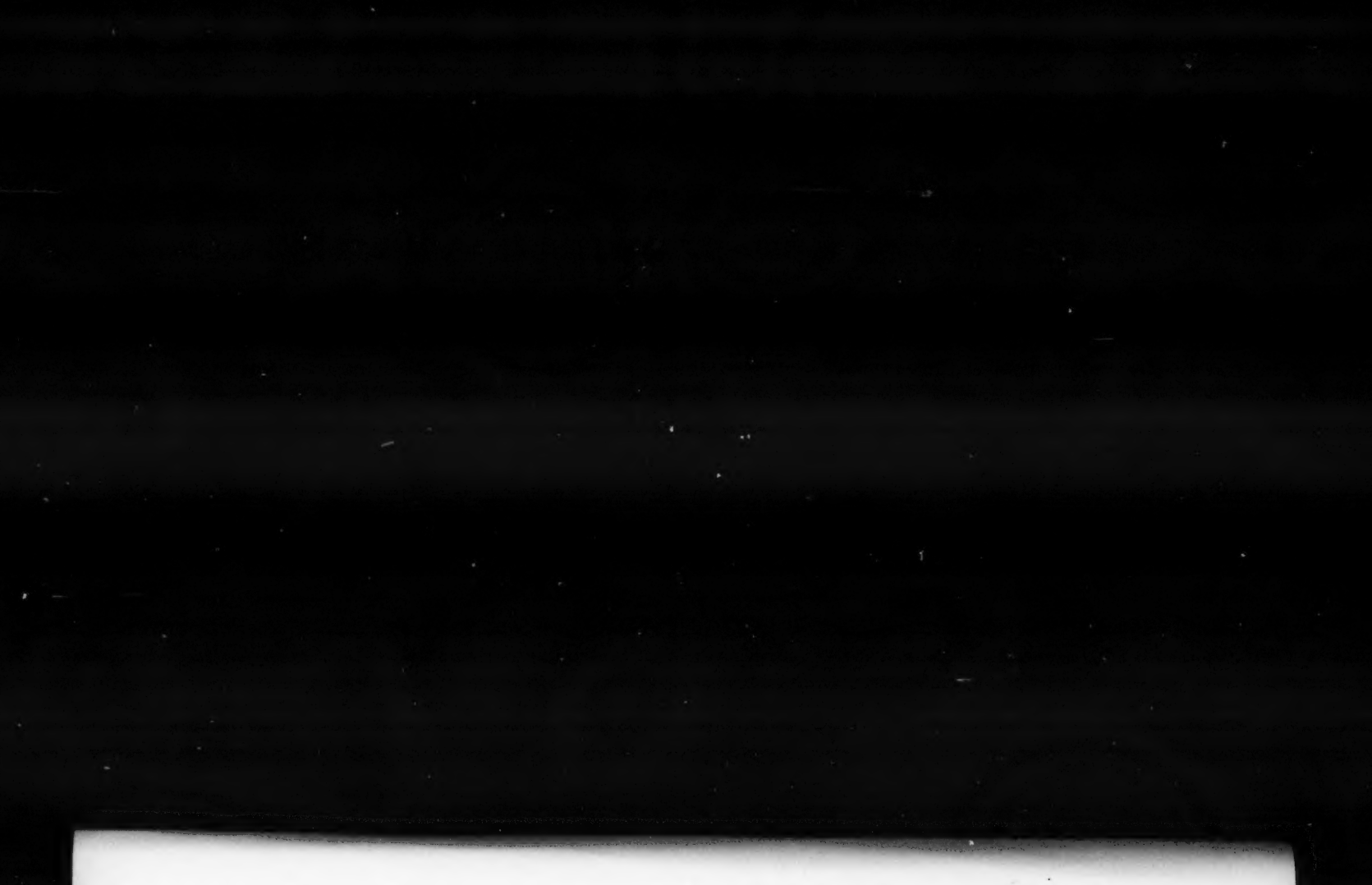
TWELVE-HOUR TWENTY-FIVE-KNOT TRIAL.

PERKINS.

This trial was begun at 6 A. M., September 29, 1910, and ended at 6 P. M. of the same day. Throughout the trial the weather was clear, cool and pleasant with smooth sea and long swell.

Three boilers (A, B and C) were used. During the trial dense black smoke issued continuously from the funnels. Seven burners were in operation in each of boilers A and B, and five in boiler C; fuel oil was not heated before burning.

The exhaust steam from all auxiliaries was led to the sixth stage of main turbines. Trap drains from galley, heating system, etc., discharged to the port main condenser; auxiliary condenser and evaporating plant were not in use.



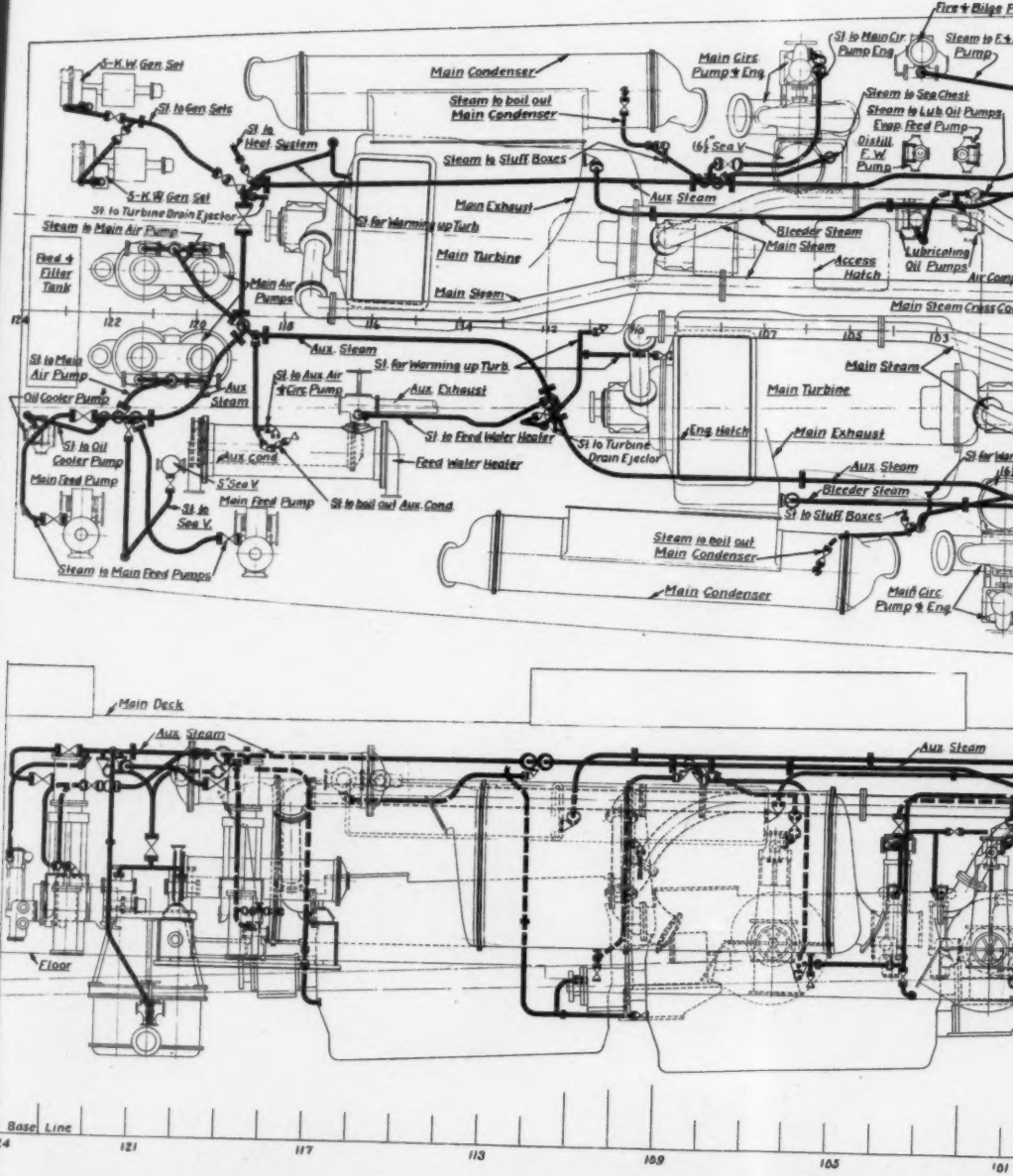
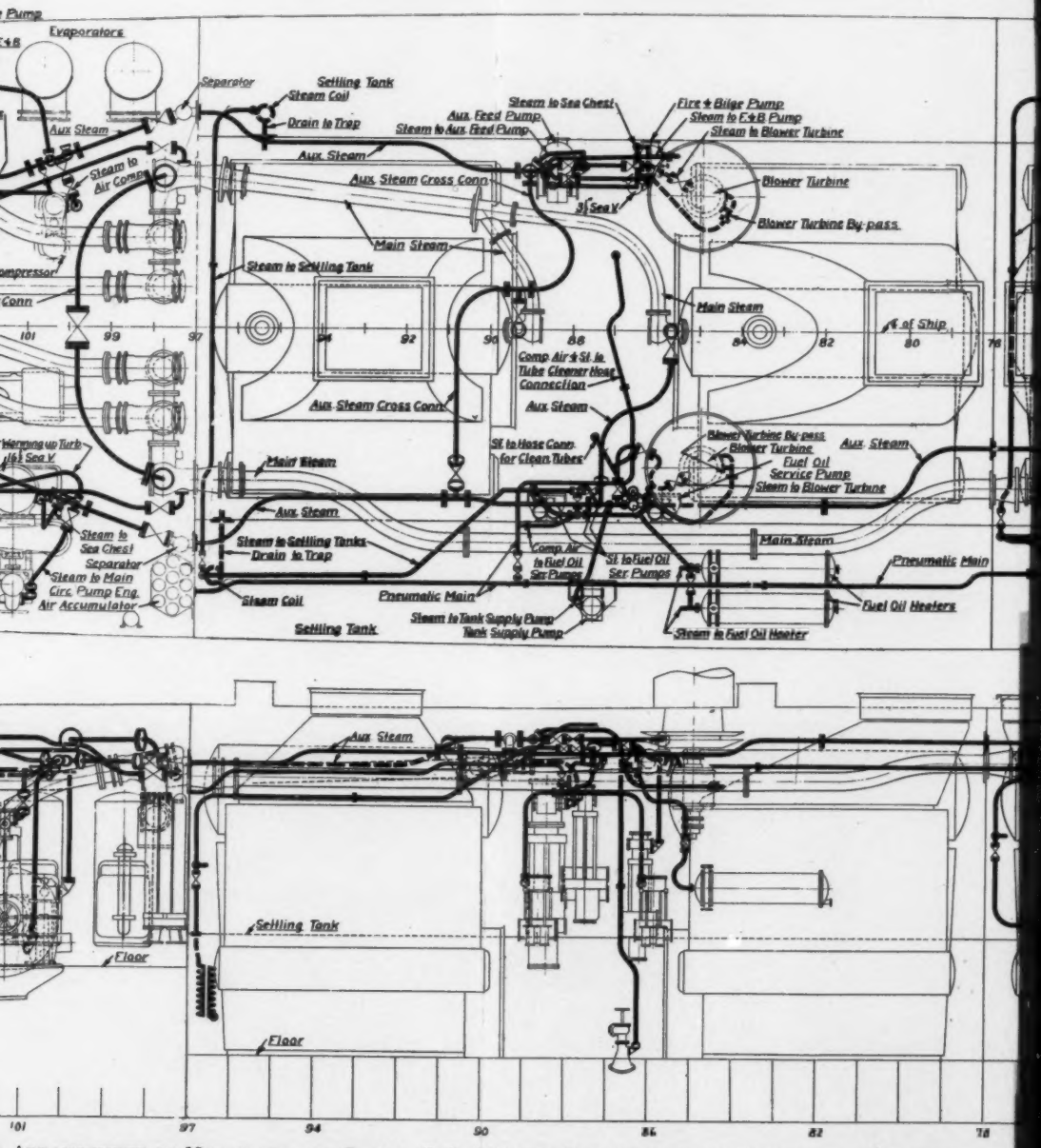


Plate. 3—General A



ARRANGEMENT OF MACHINERY AND PIPING, U. S. TORPEDO-BOAT DESTROYERS "PERKINS" AND "STERETT."

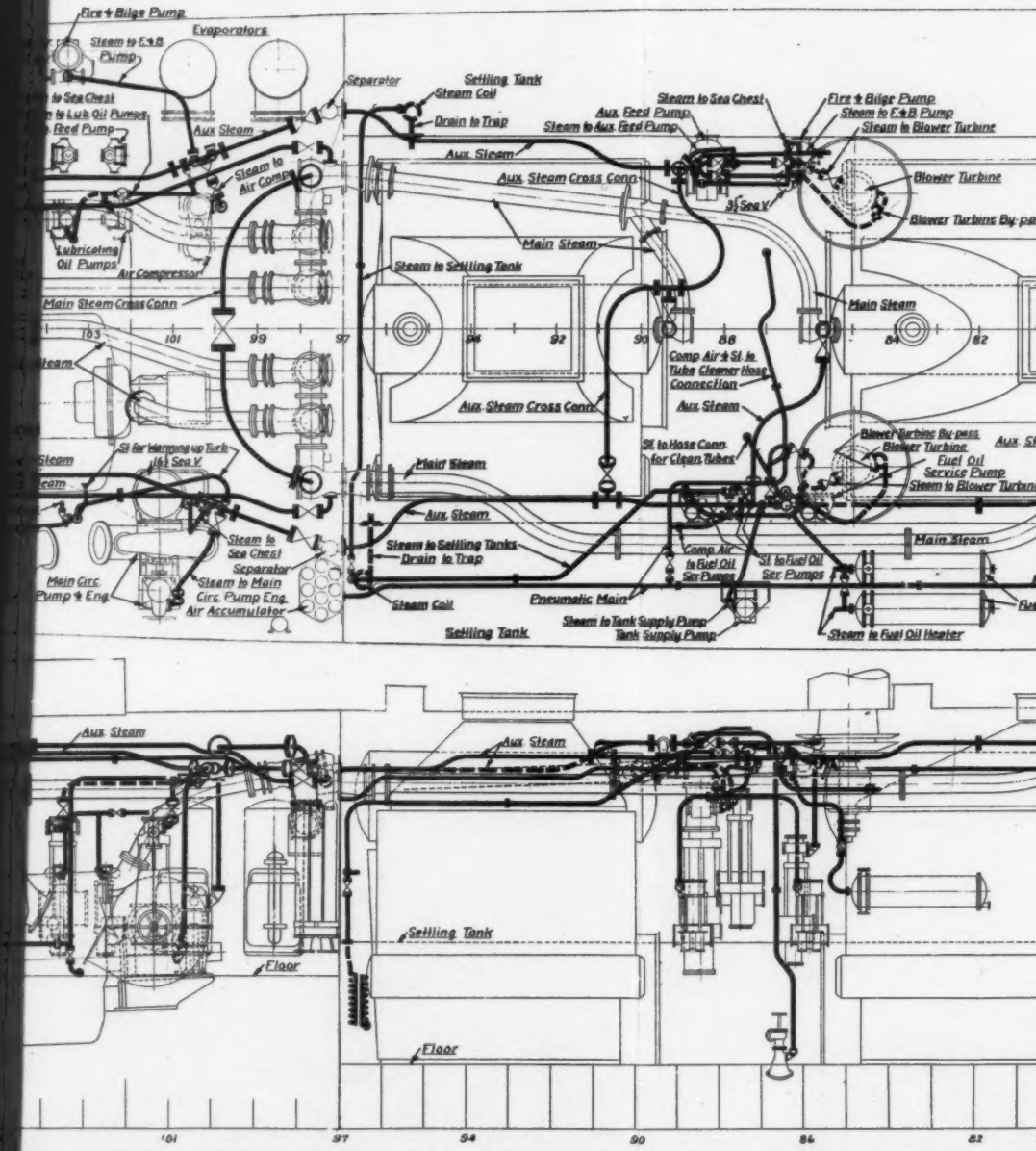
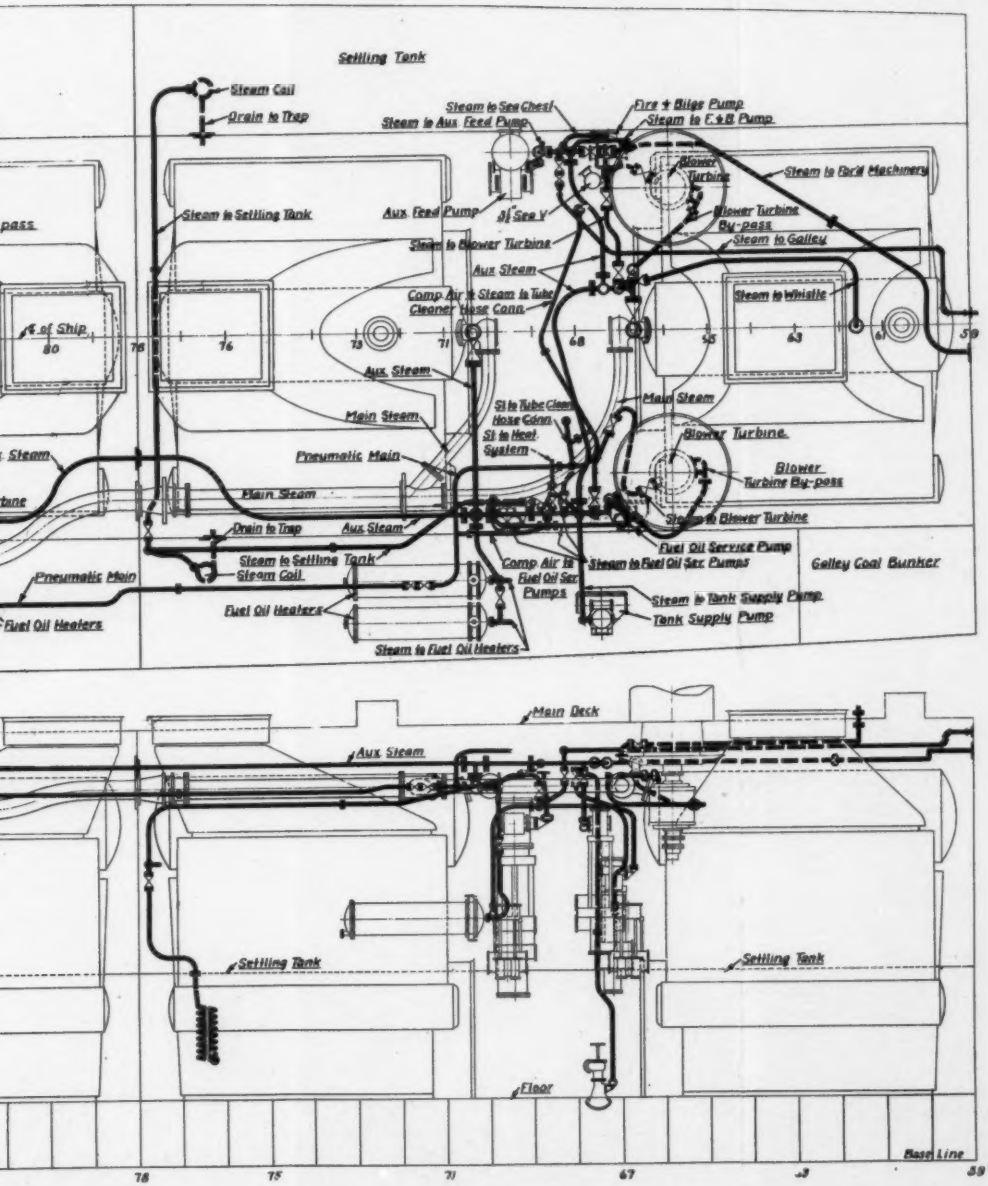


Fig. 3—General Arrangement of Machinery and Piping, U. S. Torpedo-Boat Destroyers "Perkins" an


Settling Tank
Steam Coil
Drain to Trap
Steam to Sea Chest
Steam to Aux. Feed Pump
Fire & Bilge Pump
Steam to F.&B. Pump
Blower Turbine
Steam to Deck Machinery
Steam to Settling Tank
Aux. Feed Pump
Sea V.
Blower Turbine By-pass
Steam to Blower Turbine
Steam to Galley
Aux. Steam
Comp Air & Steam to Tube Cleaner Hose Conn.
Steam to Whistle
C of Ship
80
78
76
73
71
68
65
63
61
59
Aux. Steam
Main Steam
Si to Tube Cleaner Hose Conn.
Si to Heat System
Pneumatic Main
Blower Turbine.
Blower Turbine By-pass
Main Steam
Steam to Blower Turbine
Drain to Trap
Aux. Steam
Fuel Oil Service Pump
Steam to Settling Tank
Steam Coil
Comp Air to Fuel Oil Ser. Pumps
Steam to Fuel Oil Ser. Pumps
Galley Coal Bunker
Pneumatic Main
Steam to Tank Supply Pump
Fuel Oil Heaters
Fuel Oil Heaters
Tank Supply Pump
Steam to Fuel Oil Heaters
Main Deck
Aux. Steam
Settling Tank
Settling Tank
Floor
Base Line
78
75
71
67
63
59

AND "STERETT."

STERETT.

This trial covered a period from 6 A. M. to 6 P. M., on November 2, 1910. The weather was fair, cool and pleasant, with light to moderate breezes and a light swell.

All boilers were under steam with four burners in each in use; fuel oil was heated before spraying.

During the trial, the exhaust from auxiliaries in operation, as well as the discharge from various traps in use, was directed as above stated on the similar trial of the *Perkins;* auxiliary condenser and evaporating plant were not used.

AUXILIARIES IN OPERATION.

Perkins.	*Sterett.*
2 main air pumps.	2 main air pumps.
2 main circulating pumps.	2 main circulating pumps.
1 feed pump.	1 feed pump.
3 forced-draft blowers.	3 forced-draft blowers.
2 forced-lubrication pumps.	1 forced-lubrication pump.
1 fire and bilge pump.	1 fire and bilge pump (sanitary).
3 fuel-oil pumps.	2 fuel-oil pumps.
1 dynamo.	1 dynamo.
1 steering engine.	1 steering engine.
	1 auxiliary feed pump (stand-by).
	1 oil-tank pump (occasionally).
	1 oil-cooling pump.

The draught taken at the beginning and end of the trials was as follows:

	Perkins.	*Sterett.*
Draught at beginning:		
Forward, feet and inches	8–10 3/16	8–09 1/8
Aft, feet and inches	8–05 1/4	8–07 1/16
Mean, feet and inches	8–08 3/32	8–08 9/16
Displacement, tons	786.0	789.3
Draught at end:		
Forward, feet and inches	7–11 7/16	7–10 1/16
Aft, feet and inches	8–07 1/16	8–07 3/16
Mean, feet and inches	8–03 9/16	8–08 9/16
Displacement, tons	730.5	723.5
at middle of trial (estimated), tons	757.0	756.5

PERFORMANCE, TWELVE-HOUR TWENTY-FIVE KNOT TRIALS.

	Perkins.	*Sterett.*
Average steam pressures :		
At boiler, gage, pounds	260.0	263.0
Engine room, main steam, gage, pounds	239.0	252.0
auxiliary steam, gage, pounds	243.0	257.0
Auxiliary exhaust, gage, pounds	3.8	5.2
On glands, gage, pounds	6.5	7.0
Starboard steam chest, gage, pounds	234.0	243.0
Port steam chest, gage, pounds	234.0	243.0
Starboard, 1st stage, gage, pounds	33.1	40.7
Port, 1st stage, gage, pounds	32.1	40.3
Number of nozzles open on H.P. steam chest	7.0	7.0
Pressure in oil system	15.0	13.8
Vacuum in condenser, starboard, inches of mercury	28.3	28.8
Vacuum in condenser, port, inches of mercury	28.1	28.8
Air pressure in fireroom, forward, inches of water	2.4	2.1
aft, inches of water	1.7	1.5
Barometer	30.3	30.4
Revolutions or double strokes per minute :		
Shaft No. 1, starboard	480.8	496.6
2, port	478.6	482.3
Average	479.7	489.5
Pumps, main air, starboard	28.9	30.3
port	28.5	30.0
circulating, starboard	146.2	120.0
port	155.8	104.0
feed, forward	...	30.6
aft	27.7	...
forced-lubrication, starboard	...	20.2
port	17.0	...
fuel-oil, forward	25.0	34.2
aft	12.4	32.8
Forced-draft blowers	1,330.0	1,515.0
Speed of ship, knots	25.2	25.22
Slip of propeller in per cent, of its own speed, based on mean pitch :		
Shaft No. 1	16.6	18.4
No. 2	16.0	15.8
Mean	16.3	17.1
Shaft horsepower :		
Shaft No. 1, starboard	3,265.0	3,535.0
No. 2, port	3,220.0	3,230.0
(*a*) Total	6,485.0	6,765.0
(*b*) From standardization curve at revolutions corresponding to trial speed	6,450.0	6,500.0

Temperatures, degrees Fahrenheit:	*Perkins.*	*Sterett.*
Injection	55.3	49.6
Discharge	80.9	79.2
Air pumps, discharge	67.7	67.9
*Feed	66.0	67.2
Outside air	58.8	51.7
Engine room, working level	92.0	89.2
Fireroom, working level	91.0	83.0
Smokepipe gases, average, degrees F	899.0	...
Fuel oil:		
Kind	Texas.	Texas.
Specific gravity	.894	.884
Pressure at burners, gage	85.3	108.0
Temperature at burners, degrees F	...	107.5
B.T.U., per pound	19,750.0	19,679.0
(*m*) Pounds per hour	8,392.0	8,926.0
Knots per ton (2,240)	6.73	6.73
Water:		
(*g*) Pounds per hour, all machinery	100,067.0	105,128.0
per S.H.P. main engines	15.43	15.54

DEDUCED DATA.

(*q*) Square feet of heating surface per S.H.P.	2.01	2.66
(*r*) Pounds of oil per S.H.P. per hour	1.29	1.31
(*u*) Cooling surface (main condensers) square feet per S.H.P.	1.48	1.41
(*y*) Heating surface used during trial	13,500.0	18,000.0
(*z*) Pounds of water evaporated per pound of oil $\frac{(g)}{(m)}$	11.92	11.77

TWELVE-HOUR SIXTEEN-KNOT TRIAL.

PERKINS.

This trial was run on September 28, 1910, and covered a period from 7:05 A. M. to 7:05 P. M. The vessel had been weighted for the twenty-five-knot trial, but owing to weather conditions it was believed inadvisable to carry out a trial at that speed, and the sixteen-knot test, without change of displacement, was undertaken instead.

At the beginning the weather—which later cleared—was foggy; the sea was comparatively smooth with a long swell,

*Feed heater not in use.

and a moderate breeze at first from the S.W., which gradually shifted to W.N.W.

Two boilers (C and D) were used with three burners in each in operation; fuel oil was not heated before being fed to the burners.

Exhaust steam from auxiliaries in use was led into the first stage of main propelling turbines. All trap drains (galley, heating system and other traps) discharged into the port main condenser; auxiliary condenser and evaporating plant were not in use.

STERETT.

The sixteen-knot trial of the *Sterett* began at 5:30 A. M. on November 3, 1910, and ended twelve hours later. The weather was overcast, cloudy and cool. At the start there was a moderate breeze from E.S.E., which gradually shifted during the trial to a moderate breeze from E.N.E. The sea was smooth with a swell from the southward. During the latter half of the trial there was a short, choppy sea from E.N.E.

Boilers C and D were in use on this trial, with two burners in each—outward burners of lower row. Fuel oil was heated before spraying to an average temperature of 138 degrees F.

The disposal of auxiliary exhaust, trap drains, etc., was the same as mentioned for the *Perkins*.

AUXILIARIES IN OPERATION.

Perkins.	*Sterett.*
2 main-air pumps.	1 main-air pump.
2 main-circulating pumps.	2 main-circulating pumps.
1 auxiliary-feed pump.	1 auxiliary-feed pump.
1 forced-draft blower.	1 forced-draft blower.
2 forced-lubrication pumps.	1 forced-lubrication pump.
1 fire and bilge pump (sanitary).	1 fire and bilge pump (sanitary).
1 fuel-oil pump.	1 fuel-oil pump.
1 dynamo.	1 dynamo.
1 steering engine.	1 steering engine.
	1 oil-cooling pump.
	1 oil-tank pump.

The draught, taken at the beginning and end of the trials, was as follows:

	Perkins.	*Sterett.*
Draught at beginning, forward, feet and inches	8–11⅛	8–08 1/16
aft, feet and inches	8–05 1/16	8–06¼
mean, feet and inches	8–08 1/32	8–07 5/32
Displacement, tons	788	776
Draught at end, forward, feet and inches	8–08 1/16	8–05⅜
aft, feet and inches	8–05⅛	8–05 9/16
mean, feet and inches	8–06 5/32	8–05 21/32
Displacement, tons	767.5	755.5
(estimated) at middle of trial	775.5	765

PERFORMANCE TWELVE-HOUR SIXTEEN-KNOT TRIALS.

Average steam pressures:	*Perkins.*	*Sterett.*
At boiler, gage, pounds	258.0	264.5
Engine room, main steam, gage, pounds	250.8	252.5
auxiliary steam, gage, pounds	248.5	256.0
Auxiliary exhaust, gage, pounds	6.8	8.6
On glands, gage, pounds	5.4	4.1
Starboard steam chest, gage, pounds	243.8	* 193.0
Port steam chest, gage, pounds	243.0	230.0
Starboard, first stage, gage, pounds	0.0	4.1
Port, first stage, gage, pounds	0.0	4.2
Number of nozzles open on H.P. steam chest	2.0	2.0
Pressure in oil system, pounds	15.0	14.5
Vacuum in condenser, starboard, inches of mercury.	28.2	27.6
port, inches of mercury	28.2	28.4
Air pressure in fireroom, in inches of water	1.2	1.4
Barometer	30.1	30.5
Revolutions, or double strokes, per minute:		
Shaft No. 1, starboard	302.36	304.21
2, port	299.96	292.28
Average	301.16	298.25
Pump, main air, starboard	28.2	...
port	28.7	34.2
main circulating, starboard	111.0	99.5
port	113.0	102.1
main feed, forward	Not in use.	Not in use.
after, auxiliary feed	12.1	10.2
forced-lubrication, starboard	...	21.6
port	16.0	...
fuel-oil, forward	...	28.9
aft	16.1	...
Forced-draft blower (one)	1,371.5	1,222.0
Speed of ship, knots	16.61	16.27

* One nozzle valve found off its seat at inspection of machinery after trial.

	Perkins.	*Sterett.*
Slip of propeller, in per cent. of its own speed, based on mean pitch:		
Shaft No. 1, starboard	12.6	14.09
No. 2, port	11.7	10.41
Mean	12.15	12.25
Shaft horsepower:		
(*a*) Total	1,544.0	1,596.0
(*b*) From standardization curve at revolutions corresponding to trial speed	1,520.0	1,400.0
Temperatures, degrees F.:		
Injection	55.5	50.0
Discharge	72.7	66.0
Air pumps, discharge	59.8	53.5
*Feed	64.0	55.4
Outside air	61.8	48.4
Engine room, working level	102.0	85.2
Fireroom working level, aft	116.0	95.0
Smokepipe gases, degrees F	526.0	514.0
Fuel oil:		
Kind	Texas.	Texas.
Specific gravity at 90 degrees F	.874	.883
Flash point, degrees F	270.0	260.0
Burners in use, each boiler	3.0	2.0
Pressure at burners, gage	68.0	158.0
Temperature at burners, degrees F	...	138.0
B.T.U. per pound	19,752.0	19,679.0
Ultimate analysis:		
Hydrogen, per cent	12.84	12.94
Carbon, per cent	86.90	86.54
Sulphur, per cent	.28	.21
(*m*) Pounds per hour	2,757.0	2,825.0
Knots per ton (2,240)	13.5	12.9
Water:		
(*g*) Pounds per hour, all machinery	33,175.0	33,364.0
per S.H.P. main engine	21.49	20.91

DEDUCED DATA.

(*q*) Square feet of heating surface per S.H.P.	5.83	5.64
(*r*) Pounds of oil per S.H.P. per hour	1.79	1.77
(*u*) Cooling surface (main condensers), square feet per S.H.P.	6.18	5.98
(*y*) Heating surface used during trial, square feet	9,000.0	9,000.0
(*z*) Pounds of water evaporated per pound of oil $\left(\frac{g}{m}\right)$	12.03	11.81

* Feed heater not in use.

MACHINERY WEIGHTS.

The total weight of the machinery of these vessels, which includes propelling turbines, boilers and their appurtenances; all fixtures in engine and firerooms, uptakes, smokepipes, evaporating and distilling plant, steam-heating apparatus, spare parts and stores, including water in boilers (steaming level), condensers, evaporators and feed tanks to two-thirds of their maximum capacity; water in pumps, pipes and stern tubes, but not including reserve-feed water in double bottoms, or spare parts not usually carried on board, was limited by the contract to 272½ tons. For excess weight over that stated, a penalty was exacted at the rate of $250 per ton, and an additional $5,000 added on account of such excess being greater than 5 per cent. of the stipulated contract weight. The total machinery weights (actual) are closely as follows:

	Perkins.	*Sterett.*
Weight of machinery, as above enumerated, including water, pounds	658,142	661,395
Weight of machinery without water, pounds	596,534	595,987

NOTES.

DEVELOPMENTS IN MARINE PROPULSION.

We are at the present time passing through a transition stage in the development of marine engineering which can hardly fail to make history. It seems a very short time since the last word appeared to have been spoken, and a definite limit reached, in the perfection to which reciprocating-engine practice could be brought. Multiple-expansion, surface-condensing, superheating and steam jacketing had all played their several parts in this, and further possible improvement had apparently been reduced to small proportions. The adoption of water-tube boilers, and to a small extent the use of oil fuel, brought about some further increase of efficiency from the fuel consumption and weight carrying point of view, but no serious claim to any large improvement could be made. With the epoch-making introduction of the steam turbine, however, a great advance was effected, and its extremely rapid growth is ample proof of the wide recognition accorded to the successful application of the rotary principle. The next step was the interposition of a steam turbine, between the exhaust of an ordinary reciprocating engine, and the condenser; a greater expansion of the steam being thereby obtained than is possible with reciprocating engines alone. The result was a combination engine, in which the exhausts from two reciprocating engines, driving wing shafts, were brought to a common turbine of large size working at low pressure and driving a center shaft as in the *Otaki, Rotorua* and *Laurentic,* not to mention the *Olympic* and *Titanic.* A great increase of steam efficiency was obtained by this method.

The great difficulty associated with the application of the Parsons expansion turbine to the propeller shaft directly is that of high revolutions, involving a loss of efficiency at the

propellers. Strenuous endeavors are being made to obtain an efficient and reliable means of gearing down the propeller shaft, which shall permit of lower propeller speeds being used. Mr. Parsons' experiment on the *Vespasian*, with direct gearing, has met with much success, and he is now supplying two sets of geared turbines to the Admiralty for torpedo-boat destroyers being built on the Clyde. The same effect is obtained by Dr. Föttinger's hydraulic gearing, which is being fitted to a steamer now under construction on the East Coast for foreign owners. An interesting feature of this design is that the water, which is heated by the transmission of so much power, is used to augment the heat of the feed. The performances of these vessels will be watched with great interest. On the other hand, by using a combination of the impulse and expansion turbines a lower speed of revolution is possible, as in the Curtis-Brown installation of H.M.S. *Bristol* and the A. E. G. German turbine engine, and it is thus hoped to regain the loss of propeller efficiency hitherto sustained with turbine engines.

In the meantime, another school of engineering thought is devoting itself to exploitation of the internal-combustion engine as the ideal marine motor. For very small sizes, engines using light spirits hold the field. For larger dimensions opinions are divided between gas engines and engines using crude oils. The invention of producer-gas plant has made it possible to consider gas engines, driving propeller shafts directly, as serious competitors in the race. Experiments on quite a large scale have been made, not, however, with altogether convincing results. An interesting case of producer-gas propulsion is afforded by a vessel approaching completion at Dumbarton. This is a 53-foot steel yacht, built for Mr. H. A. Mavor, of Glasgow. A gas engine using producer gas drives a dynamo, or generator, which in turn supplies power to an electric motor directly connected to the propeller shaft. The revolutions are thus brought down to about 350 per minute, with which speed a good propeller efficiency should be attainable. This little vessel represents generally the suggested application of electricity to the problem of marine propulsion.

Of more serious import, however, is the rapid development taking place in the growth of the oil engine proper. Until quite recently, the introduction of comparatively small powers for fishing boats has attracted most attention. But now the constant-pressure or Diesel type engine is rapidly making the development of large powers from crude oils possible in marine engines. So far as our own columns have shown, this engine has received more attention abroad than at home. But there is now to be built on the Clyde, at Messrs. Barclay, Curle & Co.'s shipyard, a vessel of 7,000 tons deadweight—very much greater than anything yet attempted—for which the builders are constructing at their engine works two sets of large internal-combustion engines of the Diesel type. The vessel is to have twin screws, which are expected to give her a speed of 12 knots. The provision for starting and stopping, and for reversing rapidly, with other minor details, which were successfully developed in the experimental vessels of smaller size, are all being incorporated in the larger vessel. Another interesting feature about this vessel is the manner in which the auxiliaries are to be worked. Hitherto steam has been generally deemed necessary, but in this case a smaller engine of the Diesel type will generate electric power for the winches, windlass and steering gear, and provide electric light for the vessel. The ship, it is reported, will have no funnels, the intention being presumably to discharge the products of combustion at the water level. In this, if the intention is carried out, she will differ from preceding Diesel-engine ships, in which the use of a funnel has been found at least desirable. Specially constructed oil tanks will take the place of coal bunkers; the usual boiler space will be available for cargo; and the oil space will only be about one-fourth of that which would be necessary for a coal-propelled vessel of the same dimensions and speed. The crude oil, on which this engine may be worked, can be obtained for about 35s. to 40s. per ton, and it is estimated by the builders that 100 tons of oil will take the vessel as far as a similar ship would be taken for the combus-

tion of 300 tons of coal. The ship is for foreign owners, although being built by a British firm.

Never, perhaps, has there been such a diversity of new departures in marine engineering within a short time as has been recently seen, and we are safe in saying that some, if not all, of these will make history. A satisfactory reflection is the thought that British engineers are still in the van when practical application of new movements is in question.—"The Engineer."

HERRICK ROTARY ENGINE.

A full description of the Herrick Rotary Engine appeared in Volume XXII, page 435, No. 2, of the JOURNAL. Recently economy trials of a 100-H.P. engine (built for experimental purposes) were carried out by Prof. F. E. Pryor, of Stevens Institute, who submitted a report covering results of the tests as follows:

The engine, rated at 100 H. P. at a speed of about 800 r.p.m. when operating with dry steam at 110 pounds pressure and with a free atmospheric exhaust, was mounted on a base and rigidly connected to a 75 kw., 120-volt, direct-current generator.

For the purpose of the test the exhaust of the engine was connected to a surface condenser that had been previously tested and found to be substantially tight. The steam pressure in the main leading to the engine from high-pressure boilers was at all times maintained at a pressure enough above the throttle pressure to insure dry steam at the engine. This resulted in furnishing a small amount of superheat, which, when computed, amounted to about an average of twelve degrees. The load from the generator during the tests was absorbed in rheostats and registered by means of calibrated voltmeters and ammeters.

Tests were made on the engine at half, three-quarter, full load and overload with varying steam pressures. As the en-

gine had a valve with fixed cut-off, the pressure entering the engine had to be varied to obtain the different loads. The pressure before the throttle having been maintained nearly constant, it was equivalent to operating the engine with a throttling governor. This resulted in higher superheat for the lower loads, as will be seen in the accompanying table of tests.

Table 1.—Tests for Horsepower and Steam Consumption of Double-Piston Type Herrick Balanced Rotary Engine.

Number of tests.	Steam pressure, pounds per square inch.		Back pressure.	Calorimeter temperature and superheat.	Revolutions per minute.	Generator readings.		Brake horsepower.	Water per hour.	Water per hour per brake horsepower.
	Before throttle.	After throttle.				Volts and amps.	Kw. output.			
			ins. Hg.	*degs. F.*					*lbs.*	*lbs.*
1	136	87.3	0.4	303 14	815	120.3 472	56.78	85.3	3,547	41.6
2	131	97	0.7	302 10	814	120.5 521	62.78	93.9	3,845	40.9
3	121*	106.3	0.0	305 9	816	121.2 579	70.17	104.8	4,146	39.6
4	129*	116.1	0.0	312 12	815	123.2 628	77.37	114.9	4,413	38.4
5	133.3	57.6	0.0	293 17	813	119.5 278	33.22	52.1	2,612	50.1
6	132	77	0.4	299 14	811	121.1 399	48.32	73.3	3,228	44.0
7	115	87.1	13.9 (Vac.)	299 10	812	122.4 554	67.81	101.1	3,623	35.8
8	126*	106.5	14.0 (Vac.)	305 9	814	121.7 674	82.03	121.7	4,238	34.8
9	129	87	0.0	302 13	816	120.2 477	57.34	86.0	3,635	42.3
10	127	67	0.0	293 12	818	118.4 356	42.15	64.9	3,007	46.3
11	130	87	19.1 (Vac.)	302 13	815	120.9 596	72.06	107.2	3,604	33.6
12	130*	106.5	20.9 (Vac.)	306 10	816	123.3 702	86.56	128.0	4,152	32.4
13	132	87	20.9 (Vac.)	302 13	817	121.5 593	72.05	107.3	3,534	32.9

* Superheat at throttle.

Duration of test = 60 minutes, except Nos. 5 and 6, which were 30 minutes.

Date of tests, November 10-16, 1910.

Table 2.—REGULATION TEST, OPERATING WITH THROTTLING GOVERNOR.

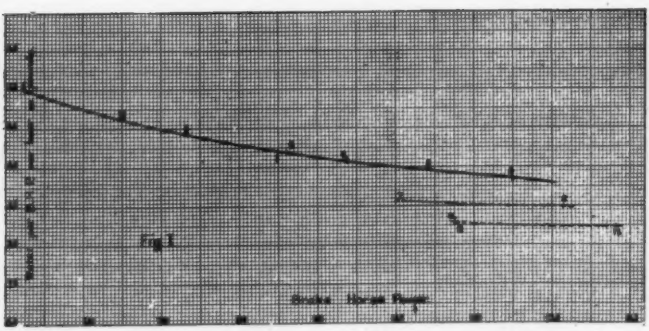
No. of trial.	Change of load.	Voltmeter readings.		Final reading after change of load.	Maximum fluctuation of voltage during change of load.
		Reading before change of load.	Reading immediately after change of load.		
	Amperes.				*Per cent.*
1	0–173	115	113	122	1.7
2	173–190	119	119	120	0.0
3	190–210	120	120	121	0.0
4	210–230	121	121	122	0.0
5	230–272	122	122	123	0.0
6	272–292	123	123	124	0.0
7	292–410	124	123	125	0.8
8	410–370	125	127	125	1.6
9	370–330	125	126	125	0.8
10	330–305	124	123	125	0.8
11	310–375	124	123	124	0.8
12	375–415	124	...	124	0.0
13	415–495	125	123	125	1.6
14	530– 0	124	140	117	12.9
15	0–270	114	109	123	4.4
16	300– 0	123	127	117	3.3
17	0–300	116	110	124	5.2
18	550– 0	121	135	118	11.6

It will be noted in the table that the temperature of the steam expanding through the calorimeter was substantially the same in all but three tests. Neglecting these variations it might be considered that dry steam was supplied to the engine at about 128 pounds pressure, and then reduced by the throttling governor to the required pressure, the engine receiving the resulting superheat due to the reduction of pressure.

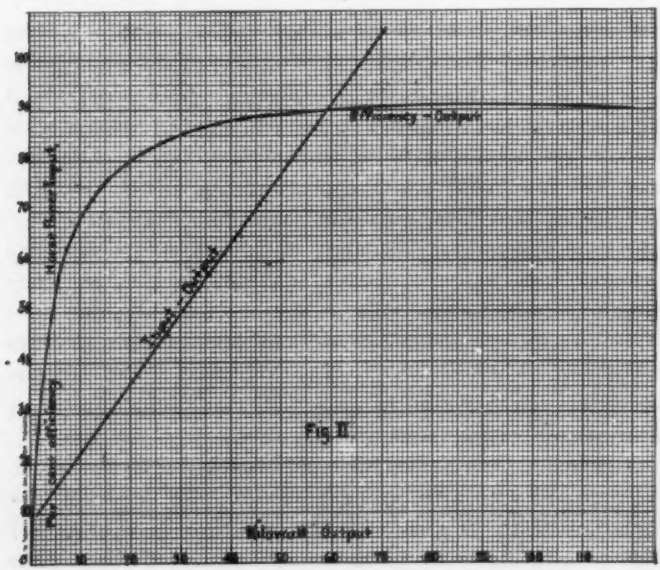
For the purpose of calculating the results of the tests the generator was calibrated for its efficiency, and an efficiency curve is plotted, Fig. 2.

Tests were also made to determine the horsepower and steam economy of the engine when it was exhausting into a vacuum. The results are shown in the same table and on the same chart as the atmospheric-exhaust test.

After making the horsepower and economy tests the engine was equipped with a throttling governor. To determine the regulation of the engine with the governor in operation I made a series of trials of throwing on and off load. The



results of these tests are shown in Table 2, the per cent. voltage fluctuation being based on variation in voltage due to the change in load. It will be noted in the table that after throwing off the load the voltage first went up and then fell to a lower point than that registered originally. This falling off of voltage is, no doubt, due to the characteristic of the generator.



The results of all tests are shown in detail in Table 2, together with a curve (Fig. 1) showing the steam used per horsepower per hour.

TURBINE TROUBLES.

Marine steam turbines of the Parsons type have now been in service for some years, so that it becomes possible to discuss the troubles to which they have shown themselves liable with some degree of fulness. Doubtless a large proportion of the earlier mishaps were such as are common to all radical departures from existing practice, until undesirable features are eliminated from the design and experience is gained in operation. However, although the design of the marine steam turbine has been considerably improved, and a long experience gained in service, it is still not altogether free from breakdowns. The greater number of accidents are seldom heard of outside a limited circle, as, naturally, those involved do not advertise such occurrences; but it is very safe to say that there are few, if any, firms manufacturing steam turbines for marine work who are without some unfortunate experiences.

In view of the thousands of blades in the Parsons turbine, the fine clearances, and varying conditions of temperature, etc., it is not surprising that blading troubles predominate, and it may be said that, apart from the blading, the possibility of breakdown is very remote.

In comparison with reciprocating engines, it must be admitted that the turbine is much freer from accidents. However, with the reciprocating engine most of the working parts are visible and accessible, and the trained eye and ear can usually locate and remedy trouble before mishap occurs. In addition, there are few conceivable breakdowns with a reciprocating engine which cannot be repaired at sea.

With the turbine the circumstances are very different. The condition of the blading and dummy rings, the most vital parts, must be largely taken on trust, and a breakdown due to blad-

ing troubles is usually of such a nature as to preclude the possibility of repair at sea.

The overhauling of a set of turbines is generally a long process, necessitating the dismantling and stowing of a great many parts, the breaking of numerous joints, and often the erection of a considerable amount of gear before the turbine casings and rotors can be lifted and the state of the blading, etc., ascertained. In the majority of cases a week at least is necessary for such an overhaul, while, with the larger installations, more than twice or even three times that period would not be too long. Thus, it is not an operation which an engineer-in-charge would be anxious to undertake. With reciprocating machinery it is the usual practice to take off the cylinder covers and examine the cylinders, pistons and valves, and also thoroughly to overhaul all the working parts at frequent intervals as a matter of routine. As explained above, the corresponding examination with the turbine system is a much more serious business, so that it is carried out at longer intervals, and the operating engineer is often entirely ignorant regarding the state of the blading, etc. The statement is sometimes made that if a turbine is not giving any trouble it is best left alone. This statement requires some modification. No doubt a turbine is best left alone so far as any interference with the dummy clearance is concerned, but an engineer-in-charge cannot see the interior of his turbines too often. In some cases an examination has revealed a state of affairs which would, sooner or later, have led to disaster, although there was no external indication of anything unusual. In the writer's opinion everything possible should be done to facilitate overhauling, as this is a distinctly weak feature of most turbine installations. The lifting gear should be as simple and direct as possible, and every effort should be made to reduce the necessary dismantling of pipe connections, etc., to a minimum.

Let us now turn our attention to the various causes of blading troubles. A large proportion is due to the rotor or casing blade tips coming into contact with the casing or rotor respec-

tively, and, although it is now the usual practice to thin the tips of the blades, the effects are only modified by this means. The disappearance of the running clearance can generally be traced to some of the various causes about to be discussed, but it is indisputable that in many cases an increase in the clearances would have avoided accident. This raises the question as to whether the usual running clearances are adequate. Theoretically, of course, reduction in blade-tip clearances is synonymous with an increase of economy, but in actual practice slight increases over those advised by the patentees do not seem materially to increase the steam consumption. Thus, one of the latest British scout cruisers achieved remarkably economical results, in spite of the fact that her clearances were all some five-thousandths of an inch greater than designed. On the other hand, a well-known firm obtained very good results with the turbines of a British battleship, and attributed this to their reducing the blade clearances. However, on attempting to repeat this performance with another naval vessel, the trials were brought to an abrupt conclusion by a most disastrous strip.

It is quite obvious that any attempt to secure superior economy by a reduction of blade-tip clearances is a most unwise proceeding, especially so on account of the parts affected being invisible and impossible of adjustment. It would seem that slight increases over the usual running clearances would greatly reduce blading risks at the expense of only a slight increase in steam consumption. This applies more especially to the turbines of naval vessels, in view of their widely varying conditions of service. In the majority of cases the disappearance of the clearances is due to distortion of the turbine casing. Any one who has not actually experimented would be struck by the curious distortion of some turbine casings under heat, and it is evident that in most cases the clearances measured cold are very much reduced when the turbine is hot. When the turbine casing is heavily ribbed most surprising distortions occur, and for this reason the castings should be as simple as possible. Deep circumferential bulb ribs should be

avoided, and the requisite strength obtained by thickening the cylinder itself. It is usual to arrange a deep longitudinal rib along the bottom of the casing to give the necessary stiffness in this direction. It is impossible to lay down any hard-and-fast rules governing the distortion of turbine casings, but the writer has observed that the distortions are of much smaller magnitude when the casings are relatively short. When the casings are abnormally long they usually develop a tendency to "hog" under heat, *i. e.*, the center tends to rise, and in extreme cases the actual amount of this distortion may make it a matter of great difficulty to make suitable allowances. Where the astern-turbine casing is incorporated with the low-pressure turbine the overhung end usually tends to rise under heat, and for this and other reasons the clearances in the astern turbine should be ample.

A very frequent cause of blading strips is local distortion of the casing produced by local temperature variations. Thus, an accident occurred to the astern turbine of a recent cross-Channel steamer on her first trip. In this case, instead of the silent blow-off pipe being led into the condenser, as is usual, it was connected to the exhaust end of the casing in way of the astern turbine, and the strip was attributed to the local distortion produced by the rush of high-pressure steam when the silent blow-off was operated. Local distortion may be the result of rapid or incomplete warming up, and in this connection it may be said that the turbine should have as long and as thorough a warming up as is possible. Any undue haste in this direction is apt to entail a heavy penalty. In the larger turbine installations special warming-up connections are fitted, and by this means more or less equable expansion is secured. In most smaller jobs, however, no such arrangements are made, and greater care is necessary. It is usual to give the rotors a turn or two with the turning gear to ensure that the temperature is equal on all sides. Neglect of this precaution when rapid warming up is essential is apt to cause severe vibration on starting, if worse trouble does not result. On the other hand, the British Admiralty do not advise the use of the

turning gear when warming up unless special arrangements are made, their reason being that if the rotor is already distorted severe damage may be done to the blading when the turning gear is operated. The best and safest practice is to make the process of warming up as long and gradual as possible.

While discussing this subject it may be noted that strips have occurred to the cruising turbines of recent torpedo-boat destroyers. The conditions under which these turbines operate are most severe, as they are frequently called into service on very short notice. The steam connection is usually on one side of the casing, so that this side is subject to a higher temperature, and local distortion must take place unless the turbine is very carefully started up. The various steam connections which enter the turbine in way of the blading are sometimes sources of trouble. The by-pass valve, which admits full-pressure steam, has been blamed for several strips in recent naval vessels. This fitting requires very judicious manipulation, and, in the writer's opinion, would be better left off, constituting, as it does, a possible source of trouble in careless hands. The closed-exhaust connections to the turbines should also be carefully handled. The branch pipes from the auxiliary-exhaust main range should be taken off at the top of the range, to preclude, as far as possible, water entering with the steam. The valves on the turbines controlling these connections should be well drained before being opened, and the opening should be gradual. Attention should be given to any leak-off connections from the glands which may enter the turbines in way of the blading, and care should be taken to clear the connecting pipes of water, and also to operate the valves in such a manner as to prevent any sudden rush of steam. There appears to be a somewhat greater tendency to produce priming with turbines than with reciprocating engines, as the turbine has a great capacity for drawing off steam. Priming is very apt to occur with a turbine installation if the stop valves on the boiler or boilers nearest the turbines are opened too wide. The writer has seen several instances of very severe priming due to

this cause, more especially with the earlier turbine vessels, and one of the Express Cunarders suffered in this way on her official trials. Any sudden rush of steam is very apt to bring water with it, and priming, once started, is very difficult to stop. Thus it is very necessary to avoid opening out suddenly any of the various valves controlling the steam supply to the turbines. For instance, the high-pressure turbine of one of the first turbine cross-Channel steamers was badly stripped on the first trip after leaving the contractor's hands, owing to the too sudden opening of the regulator.

When admitting steam to the astern turbine it will be evident that gradual opening out is especially advisable. On account of the small number of rows of blades in this turbine, a sudden opening out is almost equivalent to connecting the boiler to the condenser. In addition, this turbine under normal circumstances is revolving in a vacuum, so that severe distortion may reasonably be expected when it is very suddenly started up. For this reason astern turbines should have ample blade clearances. A case in point came under the writer's notice some time ago. A large turbine steamer was in mid-ocean, when " full speed astern" was rung down to the engine room. The order was promptly obeyed—too promptly, as subsequent events proved. For on opening up the turbines afterwards it was found that the blades in the last expansions of the low-pressure turbines were almost closed up, owing to the rush of water brought over with the steam when the astern turbine was suddenly started.

There can be very little doubt that the presence of water brought over with the steam has been responsible for several obscure cases of blade stripping, as a rush of water may very conceivably bend blades or break binding strips, and so cause disaster, and the opinion is gaining ground that the adoption of steam separators would in many cases be advisable.

When trouble arises from the mechanical construction or attachment of the blading itself it can usually be traced to some fault in connection with the lacing or binding wire, generally due to defective brazing. Segmental blading is now

largely used, and while the adoption of this system no doubt facilitates blading operations, there seems to be an impression in some quarters that there is more scope for inferior workmanship with this style than with the individual method.

Whatever method is adopted, the best workmanship and most careful inspection is essential if a sound job is to result. A recent improvement to the tools for caulking the packing pieces consists of a raised portion or island in the center of the tool, which ensures much sounder caulking with the heavier blading sections, and also leaves a distinct impression which is of considerable assistance when inspecting.

In the early days of turbine construction a large proportion of the mishaps could be directly attributed to defective design. Thus, the rotors were very crude affairs compared with the designs of today. The usual construction was to make the drum of boiler plate fitted with an internal butt-strap at the joint and stiffened with stays screwed through from side to side. In the larger sizes additional stiffness was secured by fitting internal rings of cast steel made in segments bolted together, the outer flange being attached to the inside of the drum by means of screwed pins riveted over. Such designs frequently gave trouble, and in more than one case the joint of the drum opened up, entirely wrecking the turbine.

Modern rotor drums are of weldless steel, and have frequently internal stiffening ribs turned out from the solid.

Other parts which gave trouble were the wheels securing the drums to the spindles or rotor shafts. They were almost invariably made of cast steel, and the arms or spokes had an unfortunate tendency to crack. This type of wheel is still frequently fitted in merchant vessels, often with the arms curved, but the risk of fracture is always there, so that the wheels for British naval vessels are now always of forged steel, which also simplifies balancing operations. They are generally of the "arm" type, the spaces between the arms being sawn and slotted out. Wheels of the "disc" or "dished-piston" type are also sometimes fitted, but this style of wheel is generally weak in a fore-and-aft direction, and compares

very unfavorably with the "arm" type in this respect, so that its use is not to be recommended. Cases have come under the writer's observation where the adoption of this style of wheel has resulted in the dummy rings being carried away. The rotors of the *Mauretania* are splendid examples of the very best design. In this case the wheels are of the piston type, but each wheel is made up of two pistons dished in opposite directions. These are of forged steel, screwed and shrunk upon the spindles, and bolted at their outer edges to each other and to an internal flange on the rotor drum, the whole constituting an immensely strong construction.

Apart from the blading, the possibility of trouble is remote, providing that the lubrication of the turbine bearings is efficient. Forced lubrication is usually fitted, and should give no trouble. There must, however, be no cessation of supply for no matter how short an interval, or the consequences will be disastrous. Thus, in the larger classes of British naval vessels the supply pipes to the bearings consist of two entirely independent systems. It is advisable to carry a low pressure of oil at the bearings, usually under 1 pound per square inch, so as to minimize leakage losses. Another point which should be noted is that the temperature of the bearings may quite safely be very much higher than is usual with reciprocating engines. It is now a common practice to fit thermometer connections to the oil wells below the turbine bearings, so that the temperatures may be noted. Instead of the forced-lubrication system, in some cases the oil is pumped to sight-feed lubricators on the bearings. These lubricators have usually a valve combined with them to regulate the flow, and this, together with the advantage of actually seeing the oil entering, constitutes a considerable improvement. The adjusting blocks, which fix the fore-and-aft position of the turbine rotors, sometimes give trouble at one particular speed at which the steam balance may not be good, but careful fitting and grinding up will overcome this trouble, unless the balance is very bad, when an alteration to the dummies will be necessary.

In conclusion, it must be admitted that the Parsons turbine

constitutes a great improvement on the reciprocating engine so far as freedom from running troubles is concerned, and, with ample blade clearances, good workmanship and intelligence in handling, it would be difficult to find its superior in this respect. —"The Engineer."

LIVE-STEAM FEED-WATER HEATERS.

Few questions about boilers have been more keenly contested than the heating of feed water by live steam. It is argued with much plausibility that it is impossible that the heating of feed water by the condensation of steam drawn from the boiler feed should entail economy. Thermo-dynamic laws confirm this impossibility; as well argue that a man can increase his store of coin by transferring it from a pocket on his left to one on his right side. The only possible reply is that of the man on the village green who, being told that he could not be put in the stocks because such an action would be illegal, said, "but I *am* in them." It is out of the question that any gain should be secured; yet there is most excellent evidence that it is secured. Mr. Kirkaldy has fitted a large number of steamers with his apparatus; and we have seen many reports, all highly favorable. Some years ago we ourselves carried out a test with a Kirkaldy heater on board the river steamer *Oriole* during a trip from London to Margate. We found that the firemen, not knowing whether the heater was on or not, very easily kept the pressure at 95 pounds with its aid, when without it they were pushed to keep it at 90 pounds. The revolutions were increased from 30 per minute to 32. The trials were so impartially and carefully carried out that they left no room for doubt. On the other hand, trials have been made with the result that no appreciable gain was secured. But these trials were not made with a Kirkaldy heater. Fresh light has now been thrown on the whole subject by Dr. A. H. Gibson, Professor of Engineering at University College, Dundee. He has carried out a series of experiments,

the results of which form the subject of a paper which he read before the Institution of Engineers and Shipbuilders in Scotland on the 24th January. Broadly stated, Dr. Gibson secured a gain of about 8 per cent. by the use of a live-steam heater. His experiments possess a peculiar interest for us, as they confirm the accuracy of more than one statement we have made concerning the manufacture of steam.

The experiments were very simple, and Dr. Gibson has not found it necessary to use mathematics in any shape. There is at the college an experimental boiler and engine. The boiler is of the locomotive type, with a barrel 3 feet 6 inches diameter, containing 47.3-inch tubes, 6 feet 10 inches long. The fire-box is 3 feet 5½ inches by 3 feet 3 inches by 4 feet 8 inches deep. There are 7½ square feet of grate. The engine is fitted with a brake drum and brake, so that the power can be altered. The gage pressure varied round about 100 pounds. Resting on two longitudinal stays in the steam space were placed three superimposed shallow tin-plate trays, connected to each other by short distance pieces. There was a tube through the bottom of each, down which the water overflowed into the next below, and lastly fell into the water in the boiler. The top dish was 9 inches, the second 14 inches, and the lowest about 20 inches in diameter. The feed water was delivered by a pipe passing through the top of the boiler vertically nearly to the bottom of dish number one. It will be seen that we have here the feed water pumped into the steam space—a system which we have often advocated. Dr. Gibson points out that better results might have been obtained if the trays had been larger; but for this the space was too restricted, and, we presume, the manhole too small. The temperature before overflow was measured by a mercury thermometer put into a pocket passing through the boiler crown, and resting on the bottom of the largest tray. The trial runs were divided into two distinct groups—one with a light load, the other with a heavy load. With the light load the temperature of the steam was 339.6 degrees F., and of the feed entering the heater 39.5 degrees to 40 degrees F. Leaving the heater it was 313.8 degrees, a gain

of 273.8 degrees. With the heavy load the figures remain nearly the same, except that the feed water, being larger in quantity, gained only 255 degrees. The experiments appear to have been made with much care, and included all that we usually find in a heat-balance account, such as the temperature of the flue gases, the measurement of the water and the coals, the dryness fraction of the steam, and so on. After carefully reviewing his figures, Dr. Gibson says: "Probably a conservative estimate would place the increased efficiency due to the use of the heater under exactly identical conditions of working as 5 per cent. in the light-power trial, and 8 per cent. in the heavy-power trial, and it would not appear unreasonable to assume that, had the heater been sufficiently large to enable the feed water to be heated up to full boiler temperature, these gains would have been increased to at least 6 and 9 per cent. respectively."

Results so remarkable obviously demand explanations. The method of raising the temperature of the feed water is quite different from that used by Mr. Kirkaldy, who mixes the steam and water in a vessel distinct from the boiler. But this does not appear to affect the results. Dr. Gibson's explanation is very ingenious, and would be convincing but for one fact which apparently tells against it, and to which we shall come in a moment. Dr. Gibson took a flat open cast-iron vessel about 8 inches diameter, and in it he put about 3 inches of water. This was heated by a gas ring, the flame temperature being 2,000 degrees F. The temperature of the water side of the plate was measured by means of a platinum-iridium thermo-couple placed in intimate contact with the plate, a shallow depression being made to receive the points of the couple. The results may be briefly stated. The difference between the temperature of the plate and that of the water gradually diminished from 70 degrees, the water being 120 degrees, and the plate 190 degrees to 0.5 degrees when the water boiled. His deductions are too long to give in detail here, but they simply amount to this: That the rate of transmission of heat being augmented by raising the temperature of the feed water

the efficiency of the heating surface is increased, and the temperature of the waste gases is diminished. This is borne out by the fact that, whereas with the heater the boiler made 7.56 pounds of steam per pound of coal, without the heater it made only 7.01 pounds, that is for the light load. With the heavy load the figures are 6.595 pounds and 6.25 pounds, respectively. The heater was obviously overpowered in the later case.

We have hinted that there is a difficulty in accepting Dr. Gibson's theory, which is that a similar gain should be obtainable by heating the feed water with waste steam or flue gasses distinct from, and in addition to, that secured on the ordinary basis, which takes no account of augmented efficiency of surface. It would, however, be very difficult, if not impossible, to say whether this action does or does not take place. In any case, Dr. Gibson's experiments place the whole question on a new footing, and provide a reasonable explanation of the phenomena of heating with live steam and the economy secured by its use.—"The Engineer."

FLUE-GAS ANALYSIS.

A summary of an investigation of the present practice in electric-railway power stations regarding flue-gas analysis was given by Mr. G. H. Relsay in a committee report presented to the American Street and Interurban Railway Engineering Association.

There are different opinions among operating engineers as to the importance of continuous records and intermittent records. Some believe in occasional analysis of the flue gases by means of the Orsat analyzer, so as to establish the conditions that give best economy, and then endeavor to keep the firing up to these conditions. Others believe that continuous and recorded analysis of the gases is necessary to obtain the best results.

Two methods of obtaining continuous analysis are used. One is the use of the sampling bottle which collects the average

gas for a day's run, and the other is by means of the automatic recorder which intermittently or constantly records the CO^2 at the time the gas is being made at the furnace. Some very good evidence has been presented against the sampling vessel, in that it does not indicate to the engineer the time in which he is getting good or poor results in his furnace. It only gives the average for the day, whereas the fireman can watch the continuous recorder and determine what condition gives the highest per cent. of CO^2 in his gases. In power stations where the load is a fluctuating one some seem to have difficulty in obtaining good CO^2 records.

It has been the opinion of some that the percentage of CO^2 could be considered as practically a direct indication of boiler efficiency. That is, a high percentage of CO^2 always meant a high equivalent of evaporation, and low percentage of CO^2 meant a low equivalent of evaporation. However, it has been proven quite essential occasionally to analyze the gases for CO, so as to guard against the likelihood of excess CO, as it has been conclusively demonstrated that the highest percentage of CO^2 does not mean the highest boiler efficiency and that a very high percentage of CO^2 may be obtained with very low boiler efficiency or a very low equivalent of evaporation. This may be on account of throttling the air supply on the furnace to such a very low point that there is not sufficient to burn the coal, thereby producing such a high percentage of CO in the flue gases as to reduce the equivalent of evaporation, and at the same time give a very high percentage of CO^2.

It is the practice with a few operating engineers to pay firemen a fixed sum with a bonus of variable amounts, depending on the per cent. of CO^2 obtained from their respective boilers while they are on duty. The following is a sample of what one company is doing in this respect: When products of combustion show CO^2 amounting to 15 per cent. for 10 hours, 30 cents extra per day pay will be given; 14 per cent., 25 cents; 13 per cent., 20 cents; 12 per cent., 15 cents; 11 per cent., 11 cents; 10 per cent., 10 cents; 9 per cent., 9 cents; 8 per cent., 8 cents; 7 per cent., 5 cents.

It has been the argument of the engineer that the extra amount paid the fireman is a small percentage of the amount the fireman is saving the company through his efforts to obtain the increase in his wages. This method is not generally practiced by operating engineers, and it appears to be doubtful whether a bonus should be paid firemen on the basis of CO^2 when there are other important elements entering into the proposition. This is particularly the case when extremely high CO^2 fails necessarily to indicate the most economical furnace operation and when it is possible for a fireman to obtain a very high percentage of CO^2 with a very low equivalent of evaporation.—"Engineering Record."

THE OIL ENGINE IN MARINE WORK.

Probably the recent publication of the fact that the Hamburg-American Company had placed an order with Messrs. Blohm & Voss for a 9,000-ton liner, which it was proposed to propel with Diesel engines, had a good deal to do with stimulating inquiries as to what could be done in this direction, but some of the more progressive naval constructors have been keenly interested in oil-driven engines for a long time. The principal reason for this may be found in the extraordinarily rapid development of the petrol motor for cars and boats, and more latterly for aeroplanes, wherein, of all machinery, the weight per horsepower requires to be a minimum, while to some extent it may also be partly attributed to a better appreciation of the work that has actually been done by submarine boats, and a fuller realization of the sizes in which their engines are being made.

The earlier attempts at applying internal-combustion engines to boats were on a very small scale, and the type of motor adopted was practically identical with that used on land; that is to say, it was a four-cycle single-acting petrol engine wherein reversibility was obtained by the use of clutches, or by firing a charge on the top of an upcoming piston after the

gaseous mixture had been cut off from the other cylinders. Obviously the former system becomes impracticable at outputs above about 400 brake horsepower, and the reversibility of the engine itself must really be regarded as a *sine qua non* for marine purposes. The most important use of petrol engines for ships has hitherto been for submarine work, the first British boat of this kind being passed into the Royal Navy as long ago as 1902. She was fitted with a four-cylinder petrol engine of 160 brake horsepower running at 340 revolutions per minute, but in the following vessels of the A class the motor was of the twelve-cylinder type, made by the Wolseley Company, and developed about 600 brake horsepower when running on the surface. In the B and C classes of submarine the engine power and number of cylinders were considerably increased, while in the D class boats the combined power on the two shafts is rather over 1,200 brake horsepower. The use of petrol in such confined spaces as are found in submarines is fraught with many dangers and difficulties, and great credit is due to the naval staffs who are responsible for their maintenance that accidents due to explosion have been of such rare occurrence. In some of the boats experiments have been made in running the engines with heavy oil, and a large measure of success has been achieved; so much so, in fact, that oil engines alone are now used for British boats. The same applies to both French and German submarines, all the more recent boats both in these and the British service being propelled by engines of the Diesel type.

It is a curious fact that after the proved success of petrol engines on land, nearly all the earlier designs for vessels to be propelled by internal-combustion engines should have embodied gas producers, and that the intention should have been to use producer gas as the primary fuel, and not oil, although it has long been recognized that, granted a suitable type of engine, the two are really interchangeable. The prominence given to schemes for introducing gas engines into ships has undoubtedly distracted a considerable amount of attention from the use of oil. The suitability of the type of fuel is naturally

one of the leading questions that must be determined, but it must be also reckoned with as weight. A good producer-gas engine will require at least twice the weight of fuel needed for the best oil engine of corresponding power; one type of fuel, however, may be more readily or cheaply obtainable. If oil is to be used, what is the best system on which to use it? It is the correct answer to this question that is now vexing the minds of marine engineers.

Two general types of engine are available—the four-cycle and the two-cycle; these may be subdivided into single and double-acting engines. Again, what is the system of ignition to be? Two forms of engine are available—one of the usual, the other of the Diesel type. It must be confessed at once that there is as yet insufficient experience available to determine this matter; probably no one type will be generally applicable. But it remains to be seen whether any type at all will find an assured place among propelling systems, or if oil engines will only be adopted for subsidiary harbor services, such as electric lighting or auxiliary use in steamboats. The wide application to marine propulsion that seems to be expected in some quarters must certainly be described as a dream of the future, and must remain so till one or two initial installations have been proved adequately reliable and commercially possible. Marine work differs from all land work; a complete breakdown in the engine room does not merely cause a tram car stoppage, or the temporary suspension of electric light such as would result from a similar accident in a power station, but it also involves risk to the lives of crews and passengers, or to many thousands of pounds worth of freight. Reliability is the primary consideration. It must be remembered, too, that a ship requires machinery for many other purposes than propulsion. Practically 20 per cent. of all the boiler power goes to feed the numerous and generally uneconomical auxiliary engines. Electric light must be provided; steering gear, bilge, ballast and sanitary pumps, turret-turning engines and winches, capstans, fans and a score of other small machines must be driven. Let us begin by small steps. In the experience in the Royal and

foreign navies of the use of oil engines for driving dynamos, it has certainly proved a partial success in units of 200 brake horsepower, but that success is only partial, and breakdowns have frequently occurred. Again, the record of the submarines is better, but imperative secrecy shrouds both their failures and successes, and in their case general suitability enforces the adoption of an oil engine, while, moreover, there are no auxiliaries of any moment. The past history of important improvements in marine propulsion deserves reflection; the case of the triple-expansion engine and the turbine, each of which took years to introduce, should not be forgotten. Much more recently it was strikingly evidenced by the service performances of the *Otaki* and *Vespasian* what economies could be effected over the usual tramp propelling machinery, but even with this incontrovertible evidence builders and owners have been extraordinarily slow in adopting the proved improvements. It is impossible to imagine that the oil engine will have better luck and wider adoption until its economies are also shown to be more than paper calculations. For small steamers—river boats, smacks, drifters and so on, it is already making progress, as the exhibition at Yarmouth shows. In their case the power is low, but if we take the case of a 3,000 horsepower cargo boat, we find the immediate problem to be solved is what type of oil engine to adopt. To what ships can one turn for comparative information, or what is the largest unit yet made? It must be confessed that the status of the oil engine in marine work is somewhat embryonic; those who have studied what has been done at sea as well as the land status of the engine may have a few views, but it is curious what a striking divergence of opinion there is. In the recent adoption of Diesel engines on the Continent we find three different types of engine—single-acting four-cycle, single-acting two-cycle, and two-cycle double-acting. In these cases the engine itself is reversible; in its earlier forms the screw was reversed by clutches, or some electrical system was adopted, neither of which can be accepted as a permanent solution of the difficulty. If, however, the type of engine is settled, the ex-

tent to which it will be adopted is the next consideration. Obviously, for safety, it would be better to begin by installing, say, twin screws of 1,000 horsepower each, rather than one unit of 2,000 horsepower; but then the weight and initial cost begin to rise. Elaborate cooling is necessary, which involves special pumping plant. Forced lubrication—never found in tramp steamers—is practically essential. Oil engines in place of the main steam engine will not obviate boilers. A large auxiliary boiler must be retained, and consequently an auxiliary condenser becomes necessary, whereas in a steam set the one main condenser would suffice. By the time that the whole installation is arranged to the satisfaction of the superintendent engineer responsible for producing results from it, it is doubtful if there will really be very much saving of weight, except in fuel carried for a given radius. For gas-driven ships this will not be greatly reduced, but a Diesel engine should not need more than 0.5 pound per brake horsepower, which materially reduces the requisite weight of fuel, and in itself represents a great gain in economy in those regions where oil fuel can be obtained. This latter is a point of paramount importance.

Granted that some form of oil engine is to be adopted, there are many mechanical points that need very thorough attention before any serious progress will be made. The statements made about battleships and cruisers being fitted with internal-combustion engines may for the time be dismissed as absurd. The largest marine oil motor at work is a submarine engine of 800 brake horsepower; larger engines have been built abroad, but are not on board a ship. The limit of size is found in the power that it is possible to produce in one cylinder; up to the present this has not exceeded about 500 brake horsepower, though 1,000 horsepower cylinder units are now being made on the Continent. Until material advances can be made on this, oil engines for high-powered craft would seem to belong to the distant future, and to be more suitable for the slower cargo vessels of 2,000 to 5,000 horsepower, wherein weight is of less vital importance. Why, however, ships'

boats are not driven by internal-combustion engines is not easy to understand. The 50-foot picket boat in the Navy is still driven by a water-tube boiler and small triple engine; abroad, the motor boat for ship work is generally accepted. Uncertainty of fuel supply abroad is the only reason that can be alleged against the introduction of such engine, and does not seem to be an adequate one.—" The Engineer."

PROPULSION BY DIESEL OIL ENGINES.

It is prophesied that owing to the phenomenal development of the Diesel engine, vessels propelled by steam engines will in a few years become obsolete. The progress of the Diesel engine is remarkable, and at the present moment ocean-going vessels are being equipped with this new mode of propulsion. Continental shipowners are fitting cargo boats now building with Diesel engines. Shipowners and shipbuilders in this country are already alive to the possibilities, and several firms are now experimenting. The change will be so enormous as to create a revolution in ship propulsion.

To the Dutch belongs the credit of building and equipping the first ocean-going vessel fitted with reversible Diesel engines. The *Vulcanus* was built by the Netherlands Shipbuilding Company, Amsterdam, for the Anglo-Saxon Petroleum Company. This vessel is 196 feet long and 37 feet wide, runs 8½ miles per hour, while at her greatest speed the forward movement can be changed into a backward movement in twelve seconds. She can load 1,000 tons of benzine in bulk on a draught of eleven feet. The vessel is intended for the trade on the coast in the Dutch East Indies to carry benzine in bulk between the producing centers of the owners. The *Vulcanus* is, according to the " Petroleum Review," equipped with a four-cycle Diesel engine, having six single-working cylinders, each 16 inches diameter, and being fed with crude Tarakan oil. When running at full speed the engine makes 180 revolutions a minute, but these can be reduced to 60. The

great advantage, however, of this system is that the motor can turn forward and back without disconnecting the shaft. This mode of propelling is very economical, both as regards fuel and hands employed. Firemen are no longer required. The engine-room staff of the *Vulcanus* consists of four men, and the whole crew does not exceed twelve men. The engine room contains an auxiliary Diesel motor of 50 horsepower for driving the air compressor, winches, pumps and electric-lighting set. Crude oil is consumed as fuel by both engines, and the main engine consumption is about 2½ tons per day of 24 hours at full speed. Regarding the accommodation, according to the "Motor Boat," from the the bows three-quarters of her length is occupied by three large tanks, which can be filled or discharged by rotary pumps, driven off the auxiliary engine. Both fore and main masts are fitted with derricks. The winches for the derricks are also worked by the auxiliary motor. Amidships there is the bridge and chart room. Just abaft the main mast is the engine room, and aft again is a space for stores. Over the engine room is a moderate-sized funnel for carrying away the exhaust gases, and attached to this is a compressed-air siren. There is no sleeping accommodation below decks, but the after end of the deckhouse is given over to the crew and the fore part to the officers. Comparing her with a steam-engined vessel of similar size, says the "Motor Boat," it will be seen that much is to be gained from her internal-combustion engines. In the first place the boiler space saved provides extra cargo space, and, secondly, a great saving is effected by the absence of stokers. Another important feature is that fuel will be available at actual cost price, as she is to be used for the conveyance of oil from the wells to distributing depots. Her owners are strong believers in the future of big motor vessels, and are constructing storage tanks of hundreds of thousands of tons capacity, and consider that a revolution in the propulsion of vessels is near. Regarding consumption, steam machinery would require, say, 2¼ pounds of coal per horsepower hour, that is about 11¼ tons per 24-hour day for a 450-I.H.P. engine, comparing with the

2.35 tons of oil consumed by the engine of *Vulcanus*. A given weight of crude oil would give five times the radius of action provided by coal and a steam engine of the same power, and an oil-fired boiler would consume about six tons of fuel per day as compared with the 2.35 tons taken by the internal-combustion engine. Wherein lies striking proof of the efficiency of the oil engine compared with steam.—"The Steamship."

AN OPERATOR'S VIEW OF THE DIESEL ENGINE FOR STATIONARY PLANTS.

It may be of interest to engineers and possibly to some owners of plants, to hear of some experiences in plants where these engines are in operation. As with all internal-combustion engines, there are many drawbacks which do not appear in a good steam plant, and where a steam engine will run under adverse conditions a Diesel will not operate at all.

Those of the three cylinder and six-cylinder type, with 16 × 24-inch cylinders, seem to be the most successful. In one plant a six-cylinder unit was installed which gave excellent service for the first year, with no repairs other than a new governor gear. It was necessary, however, to spend about ten hours every week in inspecting and taking up connections and various other parts. One thing that made this type of engine successful in that plant was that there were both steam and water power besides, so that it was not necessary to hire any more help, as the firemen could be used when the steam was not required, and when there was plenty of water power available ample opportunity was given to thoroughly overhaul the oil engine and get it ready to give good service when needed. Moreover, when steam was depended on alone for auxiliary power it was necessary to keep a boiler in service for two or three months, in case of emergency, and this standby loss was saved by the Diesel engine. In this class of plant the Diesel engine can be used to marked advantage and with great economy.

In a plant where the whole load must be carried by the Diesel engine, however, we have a far different proposition. In one such plant where there were two units, one new and the other about three years old, it was necessary to spend all of Saturday afternoon, when part of the factory was shut down, and nearly all of Sunday in adjusting wristpin and crankpin boxes, inspecting parts and the condition of the crank-case oil. This was regular routine work, and in case of regrinding valves, renewing the oil or anything extra, just so much more time was required. The fuel valves gave trouble by breaking at the points, and there was a great deal of wear due to the jar and shock of the six reciprocating systems working at such high pressures.

The governors gave a great deal of trouble in the way of unsteady running, so that it was necessary to replace them on both the old and the new units.

The first cylinder on the older engine after being overhauled gave great trouble by pounding at every second stroke, even when the clearance, as measured by compressing a lead wire, showed one-eighth of an inch, as required. Finally it was found that the piston head bulged upward by the heat to such an extent that when all the parts were hot it touched the cylinder head.

The cam-shaft gears of this unit were also badly worn, making new gears necessary. One cylinder of this and one of the newer engine were found to be worn from 0.045 to 0.095 of an inch large in places. The rings after a year's run became so badly carbonized as to require new ones to be substituted, and the wear of the wristpin boxes made either new boxes or shims on the wedges necessary in a short time.

Many of the published performances of Diesel engines have been those where the load factor is 25 to 35 per cent. The engines will run very steadily with such a load, because the heating and wear of the parts are not comparable to the heating and wear that occur with a load factor of 75 to 90 per cent. The load factor was the difficulty in the last mentioned plant; the management wished to carry full load all the time, and

sometimes an overload, and it was disappointing to find that the cylinders must be made tight before such a heavy load could be carried. To this end the pistons were fitted with two more rings, which made a great improvement, but in a short time it will be necessary to rebore the cylinders.

When it is necessary to remove a cylinder head about all the valve gear and other parts related to the cylinder must come with it, which will consume no small amount of time, and everything must be replaced with the greatest care or the result will be very unsatisfacory.

The clearance must be maintained just right or the proper compression temperature to ignite the oil will not be attained.

Although the regulation is said to be equal to that of steam engines, it cannot equal that of a modern automatic cutoff or Corliss engine; regulation within 2 per cent. has not been shown in actual practice, 4 per cent. being considered good.

The crank-case oil is also a source of trouble, and it is not surprising to find that the oil has disappeared after a hard week's run. Sometimes it is so thick that the bearings must be very warm before it will flow at all. The air compressor is a very important part of the plant, and it must be capable of hard, continuous service, for any drop in pressure during a heavy load means a shutdown. Careless selection of a compressor will entail a great deal of trouble, for there are some designs that are very unreliable.

The fuel economy is very high and in plants where steam is not needed for other purposes than power there is a great saving in fuel. In the matter of labor, there is very little difference from a steam plant; it is customary to pay higher wages to Diesel engineers than to engineers in steam plants of like power, and, while it is a simple matter to operate the engine, repairing requires plenty of help.

The first cost is excessive, and unless a plant has a spare unit it cannot give the service that a steam plant would, and it is a question whether the engine would not have to be replaced after ten years of service.

In writing the foregoing, it is not my intention to make any

unfair statement; my experiences with the Diesel engine have been exactly as described, and I have said nothing based on rumor or hearsay. Furthermore, I am convinced that this type of engine should, after some further experience and improvement, prove very successful and capable of continuous service at full loads.—G. H. KIMBALL in "Power."

GAS POWER.

In a well-equipped steam plant, operating condensing and containing all the refinements which tend toward economy, the ultimate efficiency depends upon the combination of a number of factors, including the boilers, engines, steam piping, condensers and other auxiliaries, as well as the coal and the care exercised in firing. With all these operating at their best, the highest thermal efficiency thus far attainable has been only about 15 per cent., and this has been possible only where a steam turbine or a compound Corliss engine was employed. Of the 85 per cent. of the energy in the coal which is unutilized, some passes up the stack both as sensible heat and as heat in unburned gases, and some is lost by radiation, but perhaps the greatest loss may be attributed to the inability of the steam engine to utilize all of the energy contained in the steam. Expressing these facts in a more concrete form, it may be said that for every ton of coal costing three dollars, only forty-five cents' worth of energy is converted into useful work.

In view of this relatively low efficiency of the steam plant it is somewhat surprising that the gas engine and gas producer have not come into more general use for medium-sized plants in this country. For with a gas engine and producer, a thermal efficiency about double that of the steam plant is attainable; that is, for a pound of a given coal twice as much useful work can be obtained. Moreover, low-grade coals may be used to advantage in a producer. The fact that the gas engine in relatively large units is a recent development, and the consequent lack of knowledge regarding it on the part of buy-

ers and operators, may account largely for the slowness with which the use of medium-size units has increased. However, the number of such units is increasing, even though not by leaps and bounds, and it behooves the steam engineer to become familiar with their operation; he cannot afford to sit idly by until they are forced upon him, but should educate himself as to all their salient features and be prepared to take advantage of an opportunity (or to meet an exigency, as the case may be), which may develop any day.—"Power and the Engineer."

MOTOR BOAT PROPULSION

The coming of the internal-combustion engine has affected in many and important ways current problems of mechanical engineering. Probably the most marked effect has been produced by its incidence upon the marine engineering as applied to propulsion of vessels of small size. Engines consuming light spirits such as petrol, oils like paraffin, or the heavier oils, are in constant use for small marine motors, and in increasing demand for larger ones.

The use of producer gas is being considered more and more as a practical proposition, and a most economical method of obtaining power afloat. Considerable success has already been attained with this form of explosive, produced from anthracite, but experiments with bituminous fuel seem to have resulted in caking, and the formation of slag. There is at present on order for a Glasgow firm a 54-foot launch with an installation of about 50 brake horsepower, using anthracite, and it will be interesting to note the result obtained. The more extensive investigations which have been and are at present being made with Diesel and other engines of large size, and the attention which is being given to the possibilities of producer gas for installations large enough to drive a battleship are outside the scope of the subject.

Speaking of small motors generally, it may be said that the speed of revolution required to obtain an efficient engine is

much higher than has been the custom with reciprocating steam engines, and it is this essential difference which has to be faced and dealt with in applying internal combustion engines to marine propulsion.

Application of these engines to land work is now comparatively simple—there, one has usually only to deal with efficiency of engine, and perhaps one unit of direct gearing. But when the power has to be made use of, through the thrust obtained by revolving a screw propeller, either in air or in water, the problem of the combined system is more complex. In the case of engines used for aviation purposes, the medium from which thrust is derived has so little weight that a very large diameter of propeller is demanded to obtain adequate propulsive force from power expended. In most cases a comparatively small disc area is adopted, causing a large amount of slip, and often a consequent low efficiency. To obviate this difficulty, some aviators are using twin propellers connected to the same engine, by chain or other gearing, thus obtaining a larger disc area, and acting upon a larger sectional area of air. A precisely similar state of matters obtains in regard to motor boats of moderate or even high speeds, short of the highest. There is the added difficulty, however, that the range of thrusts to be developed is much more extended, because of the varieties of size and design, and the widely varying purposes to which motor boats are now applied.

In examining reliable data which have been made public from trials of motor vessels by those who have experience of them, one is struck by the paucity of information available as to dimensions and performance of their propellers. In a list of motor-boat performances recently published by Mr. Linton Hope, comprising over 120 trials, the dimensions of only eleven propellers are given, and these are applied to such varying conditions, and with probably such varying efficiency, that not much can be gleaned from them to aid in formulating general principles.

This state of matters is quite usual, and the reason is not difficult to guess at. In motor boats, as in turbine-driven

steamers, the engine and the propeller are opposed to one another in the question of revolutions, the engine usually wanting more and the propeller wanting less for its best efficiency. This naturally results in a compromise by designers, with varying degrees of success. Information so obtained is a valuable commercial asset, and it is not altogether to be expected that the results should at once be made public, to become the property of competitors in the business of motor-boat construction. With a high-running internal-combustion engine it is very essential to obtain the designed number of revolutions, in order to develop the designed brake horsepower, so that in a large number of cases the propeller is entirely sacrificed to the engine.

This feature is not surprising, for in motor boats of moderate speed the whole amount of fuel used is not large, and a little more or less does not usually count for much in the owner's estimation. Loss of efficiency at the propeller is not nearly so obvious a matter as failure to develop sufficient power to obtain the expected speed; and so speed of boat, however attained, is the chief end aimed at.

In many cases the propeller is a reversible one—that is to say, the blades are of the feathering type, and pitch of blades may thus be altered at will from within the boat. This has much to recommend it if a strong practical pattern be adopted. It gives a rapid means of stopping the way of the boat when the pitch is reversed, and its effectiveness in this particular is surprising. Perhaps more important is the fact that the pitch for driving ahead is elastic to the extent that it may be changed to a finer or coarser angle until the required number of revolutions be obtained at the engine. Obviously this type of propeller is most convenient to the builders, and it has the still further advantage that in a head sea, or against a heavy head wind, a finer pitch may be used than for smooth-water conditions, if the normal number of revolutions have become reduced by the added resistance and loss of speed of the boat. There is, however, the drawback that the driving surface can only be made to suit one particular pitch, and there may thus

be a loss of efficiency due to variable pitch if the expected pitch do not produce the required number of revolutions.

The speed of revolutions, however, is usually high, and if the diameter be adequate the pitch will probably be fine, and the curvature of the blades not great, so that in such a case the loss due to incorrect helical surface is not perhaps very serious. Where the pitch ratio is unavoidably large, or designedly so, it may have a much more deleterious effect on efficiency.

The root flanges of reversible blades should be made very substantial, so that they may not wear unduly under the stresses they have to withstand. This involves using a large hub or boss, which is a further cause of diminished efficiency, especially where diameter is already limited to a moderate amount. These losses have to be reckoned with, but may be considered subsidiary to the great convenience of being able to humor the revolutions of the engine.

Assuming, then, that we may take advantage of this variable-pitch factor, what are the considerations governing the other dimensions of the propeller?

The primary difficulty—and it is almost an insuperable one—in the way of accurate generalization, is that of extreme variation of speed and power, when considered in relation to the lengths of the vessels which have to be dealt with and the varied uses to which they are put.

We may broadly class motor boats into three groups. First, perhaps, in importance is the rapidly-growing and developing number of boats applied to purely commercial purposes. Secondly, we may class together those boats used solely for pleasure cruising; and lastly, the extraordinary types which have been evolved to attain very high speeds on small displacements, and which are almost wholly devoted to purposes of sport. This last group now includes a departure which has the quality of being comparatively new and almost untried, namely, those boats which have been designated hydroplanes, and which are now the subject of much expensive experiment and interesting discussion. It has yet to be fully demonstrated that hyrdo-

planes will give better speed results than the best forms of motor boat of the same displacement, and of correct design for the speed attained and power carried. There is another group appearing on the horizon which is likely to become more and more important in the near future. This is the class of small boats being applied to the uses of the Navy for tenders to destroyers, vedette and picket boats, &c. It can only be a mat-

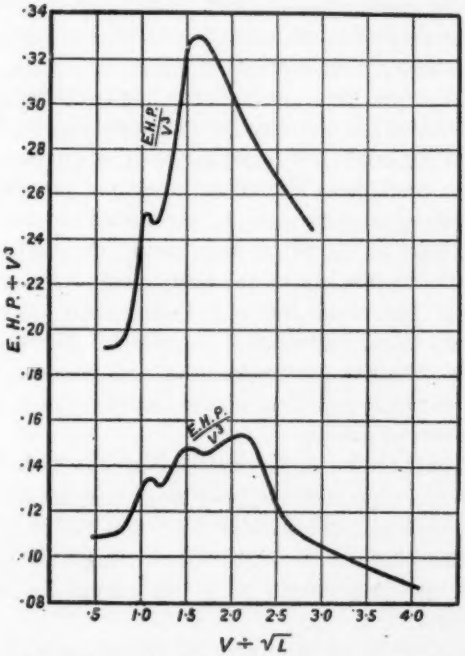


Fig. 1.—POWER CURVE OF HIGH-SPEED BOAT.

ter of time and development to accomplish the growth of this section. The new destroyers built for the Brazilian Navy by Messrs. Yarrow & Co., are each supplied with a small motor dinghy, and the British Admiralty are beginning to adapt the motor boat to the uses of our naval service. The new destroyers are all to be supplied with motor pinnaces.

Glancing over these groups, it is at once seen how wide are

the limits required to cover them. The commercial section alone, comprising as it does such an important element as the fishing industry, is wide enough to demand great elasticity of treatment. Before there was a single fishing boat in Britain equipped with a motor engine there were from 1,000 to 1,500 boats so fitted on the other side of the North Sea. An officer, specially despatched by the Japanese Government, recently investigated the application of the motor engine to fishing purposes in Northern Europe, with the result that the principle is now in process of application in Japan.

In 1901, the first large drifter was fitted with an auxiliary motor in Britain, and the example has been rapidly followed. Existing sailing boats are being supplied with motors, some having their stern posts altered to house the propeller shaft, and some simply carrying a single two-bladed propeller on one side of the boat as far aft as convenient, the shafting being inclined to the middle line of the boat to suit. Altogether the application of the marine motor to fishing craft has been, and still continues to be, increasingly successful. There are drifters, trawlers, luggers, oyster dredgers and others in the fishing-boat class alone, and these do not nearly exhaust the limits of the commercial group.

Included in it are boats for such varied uses as ferry boats, barges, lighters, tugs, coasters, pilot boats, hospital boats, boats for laying moorings, fast dispatch boats and others. It will be seen that this class itself exhibits wide ranges of speed and dimensions—possibly almost all if we except the very fast sporting types.

It is said that on the canals of the Netherlands alone there are 3,000 motor barges, while on our own canals there is not probably 1 per cent. of that number. In France and Germany the motor engine has even been applied to chain haulage on canals. This method should be very simple and economical, there being no complication due to application of the power through a screw propeller.

When we consider that the motor engine is required to apply its power in all these types of vessel, from boats having

small speed through the water and large thrust, such as tugs and canal barges, to those having an exceedingly small resistance and an abnormally high speed, such as racing boats and hydroplanes, it is apparent that the task of formulating a simple universal formula, even of an empirical nature, is well nigh impossible. From a speed point of view alone, it will be seen how much more difficult the problem is than it is for ordinary large steamships, although, of course, in the present state of development, not comparable in importance.

Suppose we take the moderate speeds of 5, 6 and 7 knots for a boat 25 feet long, and compare them with corresponding speeds for ships of varying length, we shall then see how wide the range really is in comparison.

	Length of vessel, in feet.							
	25	100	200	300	400	500	600	700
Speeds in knots.	5	10	14.14	17.32	20	22.36	24.48	26.46
	6	12	16.97	20.78	24	26.84	29.38	31.75
	7	14	19.8	24.26	28	31.3	34.3	37.04

In the early types of motor boats it was natural that they should be more or less of ordinary shipshape form, but it very soon became apparent to designers that this procedure would not do for any but the slow-speed boats, and that a different type must be adopted. The above comparison indicates the necessity for this. Consideration of the actual modifications of hull form now adopted, and their differences, must be passed by here, except to examine generally the manner in which the power varies in terms of speed generally.

Analysis of performance of vessels is often made on what is sometimes called their speed-length ratios; that is to say, the relation which their speeds in knots bear to the square roots of their lengths in feet. This is represented by $\frac{V}{\sqrt{L}}$. This figure, for even fast steamers, lies generally between 1 and 1.3 or 1.4. Now, when it is considered that for fast motor boats the

speed-length ratio for maximum speed reached is often between 3 and 4, and sometimes higher, whereas for even the fastest types of torpedo-boat destroyers it is only in the region of 2, we again see from a speed point of view how various the aspects of the propeller problem may be.

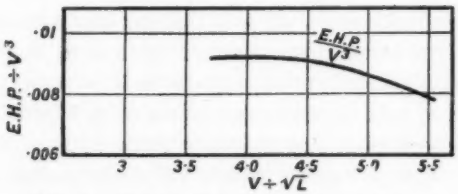


Fig. 2.—Power Curve of Racing Boat.

Not only is the actual speed high, but the rate of rise of the power curve for these speeds is by no means regular. Fig. 1 represents typical curves for high-speed boats plotted on a base of $\frac{V}{\sqrt{L}}$ and ordinates of $\frac{E.H.P.}{V^3}$. Fig. 2 is a piece of the same kind of curve for the highest speeds of a typical racer, of extreme wedge-shaped form, the water planes being almost in the form of triangles, whose apices are, of course, at the bow. These show how different the form of the curve becomes after the "worst speed" has been passed, and the impossibility of framing a simple formula to suit all parts of it.

It is fortunate that the rate of increase does not go on rising from the lower speeds continuously. A speed is reached where the rise ceases and becomes a fall. It is this feature that makes the high speeds attained by high-powered racing boats at all possible.

The critical "worst speed" has been stated in terms of displacement to be 9 times the 1/6th power of the displacement. Thus the "worst speed" for a ship of 300 tons displacement would be 23.3 knots. Obviously, however, this must vary with the length and block coefficient. The shorter the boat the lower is the critical point, and *vice versa*.

If we endeavor to build calculations upon such data as are

available we are again faced with the difficulty that, even where propeller dimensions and speed of boat are given, it is extremely rare that any information is forthcoming as to actual efficiency attained, and hence there is no indication, other than the attained speed, of the special suitability or unsuitability of the particular propeller under consideration, for its work.

It is more than probable that efficiencies which are obtained vary widely, for disc coefficients and pitch ratios differ greatly in the various examples one may examine. It is to be feared that in a great many cases the diameter adopted is inadequate for the work expected; it is, at least, a more common error than to overscrew a boat.

Very often small diameter is unavoidable owing to limited draught and necessarily high revolutions of engine, but it is sometimes due to the fear that many designers have of putting in a diameter which will involve a fine pitch ratio.

If a propeller be expected to run with only a moderate amount of slip, say, in the region of 20 to 25 per cent., it is quite proper to keep the pitch ratio at least as near 1 as possible; but where, owing to high revolutions of engine and slow speed of boat, the slip percentage is large in amount, as, for instance, in tugs and barges, it is not only permissible, but may also be beneficial, to use a fine pitch and a larger diameter. It is always difficult to estimate precisely where increase of diameter or increase of slip will exact the greater penalty, and it is this difficulty all along the range of propeller conditions for motor boats, which it is most necessary to provide for.

Fig. 3 is a curve of maximum efficiencies in terms of real slip ratio as deduced from model experiments by Froude. It will be seen how much lower the pitch ratio may advantageously be made as the slip ratio increases. It is probable that this fact is not adequately appreciated by some designers of propellers for slow motor boats which are used for heavy work.

It should be remembered, however, that, although the maximum efficiency with the particular slip expected may be at-

tained with low pitch ratio if the slip be great, it is also true that the great slip has to be paid for, and the maximum efficiency itself in such a case is necessarily low. The point is, that where large slip is unavoidable owing to the thrust required, then fine pitch ratios may be adopted, and thus a larger diameter obtained. A judicious balance of these values should result in an increase of efficiency over that of a smaller diameter adopted to give a larger pitch ratio.

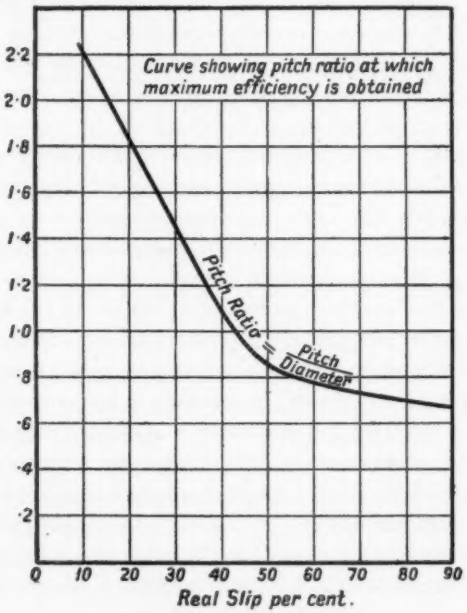


Fig. 3.—EFFICIENCY CURVE.

It should also be noted that for large ships, a considerable change of pitch ratio above that at which maximum efficiency is attained may be made with little loss, the efficiency curves being flatter than they are for the larger and more usual slips.

Attempts have been made to surmount the difficulty of inadequate diameter by fitting twin screws running at slower revolutions, both being driven by chain or spur-wheel reducing

gearing from one fast-running engine. This gives an excellent result from a speed and efficiency point of view, but has not yet been altogether satisfactory from the practical side.

It is necessary to revolve the screws in opposite directions, *i. e.*, outward on top, in order that the best results may be obtained; this involves the fitting of a wheel between the engine and one of the shafts.

Whether the gear be chain or spur wheel, it is almost inevitable that in time it will develop slap, and consequently become noisy. Apart from this, there can be no doubt that the method is a good one, and if a suitable simple gear can be devised, able to stand the stresses and to run noiselessly, with little friction, there is great attraction in it from propeller considerations for moderate-speed boats. But it is perhaps sufficient to confine our attention to the case of a single propeller driven direct by the engine shafting.

Availing ourselves of the change of pitch possible by using a reversible propeller, we want to settle upon the diameter suitable for the conditions. From an analysis of the results obtained from a number of successful boats of varying dimensions, but all of moderate speed, we may deduce a formula—empirical it is true, and only applicable to ordinary types of motor boats—from which power and length of boat are eliminated. This is due to the fact that for these boats the displacements for similar lengths approximately correspond.

The formula becomes:

$$\text{Diameter in feet} = \sqrt{\frac{(\text{Displacement in tons})^{\frac{2}{3}} \times 600}{\text{Revolutions per minute}}}$$

The diameter thus obtained is fairly adequate for boats of moderate displacement, where the weight carried is only hull and machinery, with a few passengers, &c., and where the revolutions are as high as usual in motor engines. If cargo be carried, or if speed is high, the diameter obtained is small for good efficiency. If, again, the hull and machinery are light, and the speed very high, the diameter is small. It has thus only a limited application, and is only put forward tentatively

as obtained from some actual examples. It is obvious that such a formula could not apply universally over the range to be covered.

Thrusts for unit area in these boats is not large, and the proportion of projected area to disc area need not be more than .3 or .35. This, again, must be qualified in the case of boats of great relative displacement and slow speed. It is often necessary to adopt small propellers here, because of the high revolutions requisite for the engine. In these cases much larger area should be used, the amount being proportionate to the reduction of diameter below the normal.

It is sometimes useful to throw back the blades upon a coned surface in order to keep clear of the hull, and so enable the length of outboard shafting to be reduced to a minimum. Where there is a considerable cut up of the profile aft at the keel a gain in the diameter possible may thus be made; but this can only be done, of course, on propellers of fixed pitch, not on those of reversible type.

The blades should always be as thin as is consistent with strength, and it is to be remembered that with high revolutions thrusts are smaller than would be the case with slow-running engines, and thus the blades may be kept fairly thin with safety.

When we come to consider propellers for high-speed racers and hydroplanes we enter upon ground which is largely unexplored. The approximate formula given is of no value here; the relative speeds for these boats are so very much greater than have hitherto been obtained that the question is better considered individually for each case. As a general rule, the propeller diameter has to be greater than the draught of the boat, which is, of course, built as light as possible. The designer must decide for himself how far below the keel he is prepared to carry his propeller. As a result, the propeller shaft is usually sloped at a considerable angle to the horizontal, and this is responsible of itself for a loss of efficiency. In hydroplanes which lift out of the water at the high speeds there is danger of the propeller breaking the surface and draw-

ing air. This has to be avoided by further increasing the slope of the shaft. A cardan shaft has been applied successfully to overcome this, but at high speeds the mechanical difficulties of the universal joint are considerable.

When it is considered that the cross of these joints has to reciprocate twice each revolution, it can be judged that the wear and tear upon them will be very severe at high speeds. Bevel gearing may be substituted, but is apt to become both noisy and a source of frictional loss after wear, and cannot be recommended.

There are not wanting those who are of opinion that the jet propeller may prove to be much more suitable for these fast speeds of advance than it has been shown to be at lower ones, where a larger sectional area of fluid had to be dealt with. This has yet to be proved practically, and it is always to be borne in mind that the surface friction of the water of the jet through its conduits is always bound to be a fruitful source of loss.

A method which has been tried on hydroplanes has the merit of novelty. It is to make the propeller an aerial one, such as is used for aviation purposes. There is something to be said for this method from the point of view of propulsive efficiency alone, but the diameter of an air propeller is so large that the necessary exposure to wind forces is against its adoption.

Having settled, first of all, what diameter is practically possible on the draught, and knowing the number of revolutions desired for the engine and the estimated speed of the boat, we may assume an apparent slip ratio of from 20 to 25 per cent., and from this the pitch can be immediately arrived at. It is then necessary to consider whether the pitch ratio on the diameter assumed is likely to develop a good efficiency. If the pitch ratio be anywhere in the region of 2, a good efficiency may be expected, as may be seen by reference to the curve on Fig. 3. Endeavor should thus be made so to balance diameter and pitch as to obtain slip ratio in the region of 20 per cent. and pitch ratio about 2. We are not hampered for

diameter so much at the extremely large speeds of advance attained by these racing types as in the more moderate speeds. With boats of suitable form the power falls off so rapidly in terms of speed after the "worst speed" is passed, and the speed of advance is so much accelerated, that quite moderate slips may be obtained. This is further helped by the fact that although the relative speed is so much greater than in heavier boats, the actual number of revolutions does not increase in the same proportion, for the engines do not require such an advance.

If we examine the value for $\frac{\text{Disc area} \times (\text{speed in knots})^3}{\text{B.H.P.}}$

for these fast types, we shall find that they have entirely different values from those of ordinary boats running below the "worst speed," because the speed element is relatively so high. Their coefficients will be found to be very much larger, and they must therefore be classed alone for comparison.

There is room for a great deal of exact measurement of power and speed on these conditions. Designers working along these lines not only differ greatly in their methods, but reserve to their own uses the information often expensively obtained. What is wanted is a complete series of experiments with vessels of which the E.H.P. and augmentation are known, so that actual efficiencies attained with propellers of varying diameter and pitch ratio may be accurately obtained. Too often the only factors in connection with motor-boat performances which receive any particular attention are revolutions per minute and speed. These, while quite necessary and quite a good means of comparison for different propellers on the same boat, give no indication whatever as to absolute efficiency of the combination, nor whether the highest speed attained with the best propeller tested is as high as might be expected from the mechanical possibilities of the design.

For any types other than boats of extremely light scantling, carrying as high powered an engine as may safely be used, the normal speed is below the "worst speed," so that any increase of power is usually very inadequately remunerated by

the return in added speed. The brake horsepower curve is steep; and while a large increase of power might mean only a small increase of speed, yet, on the other hand, at these speeds a considerable reduction of power is also necessary to produce a sensible reduction of speed. Hence we may have boats with differing efficiencies, and yet not having widely divergent corresponding speeds.

It is quite different in the racing types. Here the increase of speed is more commensurate with the increase of power, as may be seen from the curve on Fig. 1. The change of speed value takes place when the first crest of the wave formation of the water is thrown aft of the hull altogether, and the boat appears to ride on the back of her own bow wave. The sudden reduction in rate of increase of the resistance is a peculiar fact of experience which has not received a completely satisfactory explanation. The change of trim is due to alternation of the pressures forward and aft caused by the dynamical conditions incidental to the motion of the boat, and the change of power increase is generally attributed to the lifting effect caused by the deflection of a considerable body of water downwards, thus causing a reduction of actual displacement in the running condition and a resultant flattening of the power curve in terms of speed.

Whatever be the full explanation of this phenomenon, it is undoubtedly the reason of the evolution of the hydroplane. These boats make use of the lifting effect of planes inclined in a fore-and-aft direction at about 3 or 4 degrees to the horizontal. They naturally reach their "worst speed" much earlier than do boats of ordinary motor-boat shape, and so more quickly reach the point where speed becomes cheaper in relation to power.

Very fast motor boats have very broad water lines, the greatest breadth being generally right aft. It is necessary to give them ample stability, that they may be safe under the peculiar dynamical conditions due to their speed and power. The upsetting effect of rapid turning with helm over is apt to prove troublesome at high speeds if provision be not made for

it. More than one unpleasant experience has been recorded in this connection. The behavior of hydroplanes is likely to be erratic in the same way, especially in broken water.

The problem of propellers for these varying conditions is thus bound to be complex. For ordinary types of boat, at ordinary speeds, with direct-driven propellers, the value of the coefficient

$$\frac{\text{Disc area in square feet} \times (\text{speed in knots})^3}{\text{Brake horsepower}}$$

is usually round about 70 or 80 if the revolutions are about 700 or 800 per minute. But, as has been shown, this coefficient varies as rapidly as does the relation of brake horsepower to the cube of the speed. If the boat be a lighter of large displacement and slow speed, or a tug, then the coefficient will be very small. When, however, we pass to the other extreme, represented by the fastest racing craft, we may find the value rising to perhaps 400 or 500. So it is impossible to enunciate a simple formula to suit all conditions. Each case must be considered on its merits. Even ordinary types give their builders some trouble to obtain the best result possible. The writer knows of one firm which generally uses reversible propellers for small boats of moderate speed, and which makes the propeller blades symmetrical both in area and in section. The center of the blade is a plane, and not a helical surface, and thus a stock of blades can be kept, so that within limits changes both of diameter and pitch may be accomplished. Efficiency of propeller is thus frankly sacrificed in order to obtain the desired revolutions, and yet probably it is better than being bound both as to diameter and pitch by a solid propeller. It at least allows the makers partially to explore the possibilities of the combination, and to find out approximately the best proportions of pitch and diameter to suit the conditions. If necessary, a correctly-proportioned solid propeller can then be made for subsequent boats of the same or similar dimensions and speed, and gains of efficiency may be expected from both changes which would follow, namely, a

smaller size of hub and blades of true helical shape. The number of blades for auxiliary motor-engine propellers is usually two, so that they may be housed as far as possible when out of use by being set in line with the stern post. The use of two blades may cause some vibration when running, but will probably give higher efficiency than three or four blades, other things being equal.

For ordinary boats the usual practice from all-round considerations is to use three blades. For reversible propellers two blades are used. The material should be bronze or gunmetal, the surface well finished and buffed up.

Care should be taken that the propeller shafting runs easily in its bearings when the boat is afloat and in running condition. It is sometimes found that boats "break" sufficiently when put into the water to make the shafting—which was quite easy when the boat was on the blocks—tighten in its bearings, and so cause considerable friction.

In conclusion, it may be said that, while it is difficult to formulate a golden rule for all conditions of speed and power which may be proposed for motor-propelled vessels, there is no reason why a correct selection of dimensions of propellers should not be made, so that the best efficiency possible under the conditions laid down may be obtained. If certain general rules be followed and certain sources of possible loss be avoided, many difficulties disappear. At the same time, as in all new work, in the last resort a wise use of experience, blended with correct appreciation of the general principles involved, is the surest guide to eventual success.—"The Engineer."

PERTINENT FEATURES RELATING TO GAS POWER.*

During the fifteen years of commercial use of the gas engine in this country abundant experience has been furnished from which may be deduced two features of importance:

1. The distinct fields of usefulness of gas engines may be

* Extracts from a paper read before the Pittsburg Railway Club.

determined definitely under any conditions, and, in general, are very well defined. Contrary to the frequent implication that gas is a direct competitor of steam power or other source of energy supply, there are unmistakable regions where a gas plant is unqualifiedly superior; and, on the other hand, there are places where it would be a positive economic disadvantage. Evidently there exists a line of division or equality but occasionally encountered, where the decision rests upon probable changes in industrial or operating conditions.

2. Gas-power machinery is less responsive to the ingenious fancy of the designer than the other well known types of station equipment, because the requisite characteristics of satisfactory operation and continuity of service may be satisfied only by a simple and effective design, from which but small deviation is feasible.

The disregard of these factors more than any other cause has been harmful to the gas-engine art. Notwithstanding this the industry has materially prospered, and, owing to the inherent high efficiency in the conversion of latent thermal energy into useful mechanical power, it will increasingly continue to hold the attention of the engineering profession and commercial world as well.

The available fuels for engine operation are enumerated below and their approximate composition and calorific values are given in Fig. 1.

1. *Natural Gas,* existing principally in western Pennsylvania, western New York, West Virginia, Ohio, Kentucky, Kansas and Louisiana districts. It is a gas of ideal quality, possessing high heat value, being free from suspended impurities and containing only a small percentage of highly inflammable constituents (hydrogen chiefly).

2. *By-product Gas,* obtained mainly from blast furnaces and coke ovens; also from oil refineries as distillate. These gases are, in their crude form, accompanied by objectionable impurities—entrained ore dust in the first two and oily vapors, lampblack and sulphurous compounds in the distillate gas—

which must be removed to a reasonable degree by cleaning and scrubbing apparatus before delivery to the engine.

3. *Artificial Gas.* As several different fuels and processes are employed in the manufacture of combustible gases, there are several kinds available, such as:

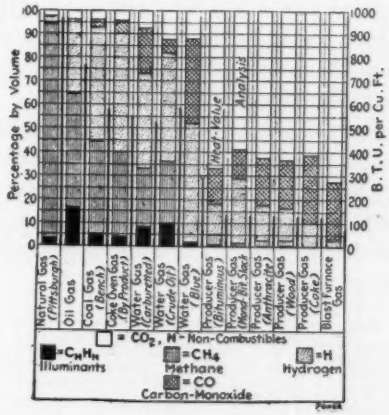


Fig. 1.—COMPOSITION AND HEAT VALUE OF FUEL GASES.

a. Illuminating Gas, produced from coal in benches or retorts by destructive distillation. This is available in practically all large cities; it is of high heat value and is a fairly clean gas. It contains a high percentage of hydrogen, which compels the use of comparatively low compression pressures to prevent pre-ignition. The enrichment essential for illuminating purposes and the added expense of distribution make the cost of this gas to the consumer ordinarily so high that its use for power purposes is restricted to very special cases.

b. City Gas, either made from the partial oxidation of coal and carburetted, or from crude oil, possesses the same limitations as illuminating gas so far as the gas engine is concerned. Blue-water gas of somewhat lower heat value and cost, is also

less satisfactory and economical than the familiar producer gas.

4. *Producer Gas. A.* The gases made from anthracite and bituminous coals present the logical solution for the use of the gas engine in the absence of by-product or natural gas at low prices.

B. Oil gas producers for power purposes have not yet attained a sufficient degree of development to warrant extended use, but where crude oil is the predominating fuel supply, some installations have been made with fair success.

Thermal Efficiencies.—The foremost characteristic of the gas engine lies in its uniformly high thermal efficiency in all sizes. This virtue is not found in either reciprocating steam engines or steam turbines; therefore the greatest utility of the gas engine, barring by-product or natural-gas supply, will occur in the smaller operations.

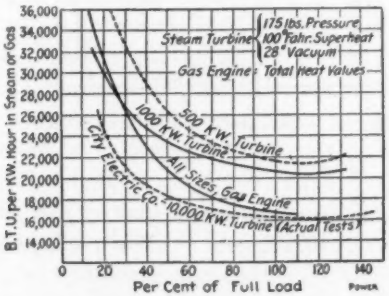


Fig. 2.—HEAT ECONOMY OF TURBINES AND GAS ENGINES.

As an illustration, Fig. 2 shows conservative heat requirements in B.t.u. per kilowatt-hour of output for turbines ranging from 500 to 10,000 kilowatts inclusive, and a typical gas-engine curve which applies to all sizes. The latter is based on the total heat of the gas and a gas quality containing a high hydrogen content (lowering the effective value, say 8 per cent.), preferably to place the two types of units on the same plane. It is evident that as the larger capacities are approached

the disparity between gas and steam units steadily vanishes, either plant at best consuming approximately two pounds of coal per kilowatt-hour. Conversely, with a decrease in size of the plant, there is a wider gap between the two types of equipment. Thus the gas plant will continue to develop a kilowatt-hour on two pounds of coal, while the steam station may use eight to twelve pounds.

Operating Economy.—The impression that gas power invariably implies a lower cost of operation is being dispelled through the critical analysis of the elements of power cost. Fuel is manifestly only a fraction of the entire expense. Customarily a division is made between fixed charges, embodying interest, depreciation, taxes and insurance, and the purely operative expense, which includes fuel, labor, oil, waste, water,

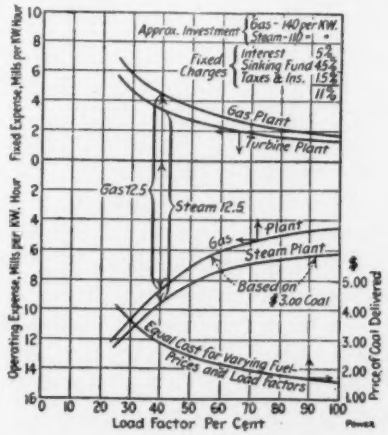


Fig. 3.—Comparative Operating Expenses.

supplies and repairs. Fig. 3 illustrates the importance of the cost constituents. It is based on plants consisting of two units and burning coal which costs $3 delivered.

The composite cost of a gas-power plant exceeds that of a high-grade steam plant by approximately 30 per cent. Such difference is quite normal in view of the character of working

in the two plants. The gas engines must withstand combustion pressures reaching 450 pounds per square inch and corresponding temperatures up to 3,000 degrees F., whereas a turbine deals with pressures of 150 to 200 pounds per square inch and temperatures in the neighborhood of 500 degrees F. Moreover, turbines are operated at relatively high speeds, and very efficient use can therefore be made of the material employed.

Labor in small gas and steam plants will not differ greatly, but the simple turbine, with its minimum of parts, places a severe handicap on the gas engine when high powers are contemplated, contributing another reason for the barrier against the use of large gas engines for central stations burning coal as fuel. Having the size and cost of the plant, together with the schedule of operating labor and supplies arranged, the decision hinges upon the probable load factor and also upon the price of fuel, where it may be subject to variation. It is an obvious fact that low-load factors work against the use of expensive installations, notwithstanding their superior efficiency.

Fortunately for the gas plant, the embarrassment of low-load factor conditions may be partially overcome by the use of a simple low-pressure turbine in conjunction with a heat-storage system supplied by the waste heat of the engine and operated to carry the peak of the load.

Available statistics indicate that the cost of repairs in well designed gas-power plants of moderate size should be about on a parity with those of high-grade steam-turbine stations. That this assertion is reasonable is evident from the fact that the greater cost of cleaning and readjustment of boiler and condenser tubes above the negligible upkeep of the producer largely offsets the increased cost of "conditioning" the gas engine above that of the turbine. Hence fixed charges were arbitrarily assumed equal in plotting Fig. 3.

The real difference in obsolescence between the two types of plant is more or less intangible at present, and, therefore, in order to avoid complication, no distinction has been made.

Table 1 is given as an example of the operating efficiency of a 550-brake horsepower producer-gas plant, including all items of expense:

Table 1.—DATA ON 550 BRAKE-HORSEPOWER GAS PLANT. MONTHLY RECORD OF OPERATING COSTS, DECEMBER, 1909. UNIT LOAD FACTOR, 89.3 PER CENT.; STATION LOAD FACTOR, 67 PER CENT.

Items.	Cents per kilowatt-hour.	
	Producer room.	Engine room.
Fuel	0.202	
Operating labor	0.120	0.036
Repairs: Labor	0.008	0.050
Material	0.000	0.002
Water	0.030	0.024
Oil and waste	...	0.010
Auxiliary power	0.050	...
Fixed expense at 15 per cent. on investment	0.127	0.254
Total	0.537	0.376
	0.913	

Auxiliary Heating.—Power generation has mainly been reckoned with as applying to central distribution, but in the machine shop, factory and related industries, the power house is subjected to the extra demand of heat supply, especially above latitude 37 degrees. The heating requirement is often improperly allowed to discount the intrinsic value of the gas plant for the reason that the waste-heat energy is not concentrated in the same convenient vehicle for transmission to the point of consumption as is the case with the non-condensing engine or turbine.

More recently gas-engine exhaust heaters have been devised which render available in the form of steam 70 per cent. of the heat of the exhaust. While this quantity represents only two pounds of steam per brake horsepower developed, it will evidently prove sufficient where the ratio of power to steam demand is low. Where the ratio of the pounds of heating steam required per brake horsepower is known, a choice of prime mover may be made as indicated by Table 2.

Table 2.—POUNDS OF STEAM PER BRAKE HORSEPOWER.

Simple automatic engine	40
Small steam turbine	30
Single-cylinder Corliss engine	28
Corliss non-condensing compound engine	22
Automatic bleeder turbine	20
Complete expansion turbine (bleeding 25 per cent. from receiver)	6
Gas engine (waste jacket and exhaust heat used in hot-water system)	5
Gas engine only, exhaust applied to steaming	2

—"Power."

SHIPS.

UNITED STATES.

Contracts for the construction of six torpedo-boat destroyers for which bids were opened on November 8th, 1910, have been awarded as follows:

No.	Name.	Contractor.	Price.
37	*Fanning*	Newport News Shipbuilding Co.	$630,000
38	*Jarvis*	New York Shipbuilding Co	640,000
39	*Henley*	Fore River Shipbuilding Co	648,700
40	*Beale*	Wm. Cramp & Sons Shipbuilding Co	654,000
41	*Jouett*	Bath Iron Works	654,500
42	*Jenkins*	Bath Iron Works	654,500

BATTLESHIP (THE *TEXAS*) NO. 35.

Bids were opened on December 1, 1910, for the construction of one first-class battleship of 27,000 tons displacement and 21 knots speed, appropriated for by Act of Congress approved June 24, 1910. A provision of the Act limited the daily service of laborers and mechanics employed in constructing the vessel to eight hours.

Bids were invited under four classes, as follows:

Class 1: Government design of hull and machinery.

Class 2: Government design of hull, with bidder's design of propelling and auxiliary machinery.

Class 3: For construction of propelling and other machinery (exclusive of electric-light plant) on the Department's design,

* The name of the present U. S. S. *Texas* has been changed to *San Marcos*.

to be delivered and installed at the Navy Yard, New York, where the hull of the vessel is to be built.

Class 4: For construction of propelling and other machinery (exclusive of electric-light plant) in accordance with bidder's plans and specifications, the completed machinery to be delivered and installed at the Navy Yard, New York, where the hull is to be built.

But one shipbuilding company submitted proposals, as follows:

Bidder.	Class of bid.	Amount of bid.	Type of machinery.	No. of shafts.	H.P. at 21-knot speed.	‡Consumption in pounds of steam, all purposes, per knot run. Speed, in knots. 12	16	18	19	20	21
Newport News Shipbldg. and Dry Dock Co.	1	$5,790,000	Parsons turbines, Dept.'s design.	4	*32,000	10,100	13,630	16,260	17,460	19,050	22,470
	2	5,830,000	Reciprocating engines.	2	†28,100	7,420	10,310	12,790	14,210	16,040	20,000
	1	5,775,000	Curtis turbines, Dept.'s design.	2	*32,000	9,350	12,800	15,270	16,500	18,100	21,500
	2	5,760,000	Curtis turbines.	2	*32,000	9,540	13,140	15,860	17,220	19,060	22,860

*Shaft horsepower. †Indicated. ‡Guaranteed steam economy at various speeds.

The contract was awarded on December 17, 1910, in accordance with the bid ($5,830,000) under Class 2, the vessel to be propelled by 4-cylinder, triple-expansion, reciprocating engines.

LAUNCHINGS.

The U. S. battleship *Arkansas* was successfully launched at the yard of the New York Shipbuilding Co., Camden, N. J., January 14, 1911. This vessel is 552 feet over all, 93 feet 2½ inches breadth on load-water-line, with a mean draught of 28 feet 6 inches. The full-load displacement is 27,243 tons; the trial displacement is 26,000 tons.

The propelling machinery consists of Parsons turbines arranged on four shafts, with an estimated total power of 28,000 shaft horsepower, and suitable to give the vessel a speed of 20½ knots at full power. The boilers are of the Babcock & Wilcox water-tube type.

MONAGHAN.

Torpedo-Boat Destroyer (No. 32) *Monaghan* was successfully launched at the works of the Newport News Shipbuilding Company on February 18, 1911. This vessel is of the same dimensions and displacement as other destroyers recently built.

U. S. SUBMARINE TORPEDO-BOAT *SEAL.*

The *Seal,* which is the first of three Lake type of submarines being built for the Government, was launched at the Newport News Shipbuilding and Dry Dock Company on February 8, 1911. This vessel has a length over all of 161 feet; extreme beam, 13 feet, and a submerged displacement of about 525 tons. The propelling machinery consists of four sets of six-cylinder gasoline engines, arranged on two shafts.

The contract stipulates a surface speed of 14 knots an hour, and a submerged speed of 9½ knots.

The following names have been assigned to submarine boats Nos. 28 to 35, inclusive:

No. 28: *Seawolf.*
No. 29: *Nautilus.*
No. 30: *Garfish.*
No. 31: *Turbot.*
No. 32: *Haddock.*
No. 33: *Cachalot.*
No. 34: *Orca.*
No. 35: *Walrus.*

U. S. S. *MARS, VULCAN* AND *HECTOR.*

U. S. S. Mars, Vulcan and Hector.—Contract for construction of these colliers—appropriated for by Act of Congress, approved May 13, 1908—was awarded to the Maryland Steel Company, Sparrows Point, Md., on October 28, 1908. These vessels, which have passed the prescribed naval trials, were built in conformity with regulations of the American Bureau of Shipping, and in accordance with rules of the United States Steamboat Inspection Service. The contract stipulated that each vessel should have approximately 7,137.3 tons' cargo carrying capacity, and prior to acceptance should successfully complete a sea trial of 24 hours' duration at an average of at least 12 knots when loaded with weights equivalent to the cargo capacity stated, in addition to the follow-

ing: 300 tons of bunker coal, 50 tons of reserve feed water, 20 tons of drinking water and 50 tons of stores.

The contract price of each vessel, fully equipped and ready for service, was $479,000.

The three vessels are alike, of steel construction, propelled by twin screws and have the following principal dimensions:

HULL.

Length between perpendiculars, feet	385-00
over all, feet	403-00
on L.W.L., feet	387-00
Beam, extreme, feet and inches	53-01½
molded, feet	53-00
on L.W.L., feet and inches	52-04
Draught (designed), mean, feet and inches	24-06

MACHINERY.

Engines:

Type, twin-screw, 3-cylinder, vertical, triple-expansion, with cranks at 120 degrees.

Diameter of cylinders:	
H.P., inches	22
I.P., inches	37½
L.P., inches	60
Stroke, inches	42
Diameter of piston rods, inches	5½

	H.P.	*I.P.*	*L.P.*
Number and type of valves	1 piston.	1 piston.	1 slide.
Diameter of valves, inches	9½	19	46 × 52½
Travel of valves, inches	8	8	8

Length of connecting rod between centers, inches	96
Ratio of connecting rod to crank	2.286

	Length.	*Diameter.*
Crank shaft (steel, built up), feet and inches	18-09½	0-12½
pins (steel), feet and inches	0-13½	0-12½
Thrust shaft (steel), feet and inches	11-00	0-11½
Line shaft (steel), feet and inches	17-03	0-11½
Propeller shaft (steel), feet and inches	24-09½	0-13

Condensers:

Type, surface, cast iron, built-in, one for each main engine.

Number of tubes (each condenser)	1,212
Diameter of tubes (outside), inch	0-00⅝
Length of tubes, feet and inches	15-07½
Thickness of tubes, B.W.G.	16
Cooling surface (each condenser), square feet	3,096.0

Propellers:

Number and material: two, built up, semi-steel hubs, bronze blades.
Number of blades (each propeller) 4
Diameter, feet and inches 14-09
Pitch, adjustable, feet and inches, from ..... 13-09 to 14-09
as set, mean, feet and inches 14-03
Area, projected, square feet 59.8
helicoidal, square feet 66.04
disc, square feet 170.87
Ratio of diameter to pitch 1.035

Boilers, main:

Number and type: four, single-ended, cylindrical, return, fire-tube.
Designed working pressure, pounds 200
Thickness of shell plates, inches ..... $1\frac{1}{64}$
Diameter (external), feet and inches ..... 15-02$\frac{1}{32}$
Length (external), feet and inches ..... 10-10$\frac{1}{2}$
Grates, length feet ..... 5$\frac{1}{2}$
width, feet ..... 3$\frac{1}{2}$
Number of furnaces (Morison corrugated), each boiler 3
Diameter of furnaces, internal, inches 44
external, inches ..... 48$\frac{1}{4}$
Total grate surface, all boilers, square feet 234.7
heating surface, all boilers, square feet 10,200
Diameter of tubes, outside, inches ..... 2$\frac{1}{4}$
Length between tube sheets, inches 86
Number of tubes (160 stay, 304 ordinary), one boiler 464
Thickness of tubes, B.W.G ..... 7 and 10
smokepipes 1
Area of smokepipe, square feet 46.25
Kind of forced draft ..... Howden.

Boiler, auxiliary:

Number and type: one, single-ended, horizontal, cylindrical, return, fire-tube.
Designed working pressure, pounds 200
Thickness of shell plates, inch ..... $00\frac{1}{16}$
Diameter (external), feet and inches ..... 8-01$\frac{1}{4}$
Length (external), feet and inches 10-06
Grate surface, square feet 22
Heating surface, square feet 622.7
Diameter of tubes, outside, inches 3
Length between tube sheets, inches 94
Number of tubes 83

Auxiliary machinery: No	Diameter cylinders, inches. Water.	Steam.	Stroke, inches.
2 air pumps attached to engines	26	...	13
2 circulating pumps, centrifugal	...	8	8
2 main feed pumps, vertical, simplex	10	14	24
1 auxiliary boiler-feed pump, vertical, duplex	2½	4½	4
4 bilge pumps attached to engines	4	...	13
1 fire and bilge pump, vertical, duplex	7½	10	12
1 sanitary pump, vertical, duplex	4½	5½	5
1 ballast pump, vertical, duplex	14	12	12
2 forced-draft blower engines, double-cylinder	...	5	4

Other auxiliaries:

Steam steerer and capstan; auxiliary condenser with combined air and circulating pumps; Remington (ammonia) ice machine (to cool 800 cubic feet volume); evaporating and distilling plant, with necessary pumps, of five tons' daily capacity of drinking water; 15 kw. dynamo plant.

Prior to the 24-hours' sea test these vessels were standardized over the Delaware Breakwater course, and information so obtained used in constructing speed-revolution curves. The following synopsis covers the important data of these trials:

	Mars.	*Vulcan.*	*Hector.*
Displacement on standardization trials, tons.	11,282	11,225	11,235
Corresponding mean draught, feet and ins...	24–08½	24–07½	24–07⅝
Mean revolutions for speed of 12 knots	86.9	85.85	85.43
Date of 24-hour trial, 1909	Aug. 18–19.	Sept. 15–16.	Oct. 14–15.
Mean displacement, tons	11,292	11,193	11,162
Average revolutions (both engines) for trial.	93.34	92.18	93.68
speed, knots per hour..	12.65	12.82	12.87
steam press. (gage) at engines, lbs.	179	182.6	196
(abs.), 1st receiver, lbs.	78.8	73.4	84.7
2d receiver, lbs.	26.4	26.0	29.2
air press. in blower duct to boilers, inches of water	2.0	...	2.0
Collective I.H.P. of main engines	3,818	3,736	3,921
Average pounds of coal used per hour	7,103	6,313	6,511

DEDUCED DATA.

	Mars.	*Vulcan.*	*Hector.*
I.H.P. (main engines only) per sq. ft. grate surface..	16.27	15.92	16.71
Pounds coal per I.H.P. per hour (main engines only)	1.86	1.69	1.66
square foot grate per hour	30.26	26.90	27.74

FRANCE.

Cruiser *Edgar Quinet:* This vessel has recently completed her official trial during which she maintained a speed of 23.92 knots per hour for three consecutive hours, thus exceeding her contract guarantee by about 1 knot. The developed I.H.P. was also exceeded by nearly 4,000.

The leading particulars of this vessel are:

Length over all, feet and inches	515–10
Extreme breadth, feet and inches	70–07
Draught, astern, feet and inches	27–04
Displacement at this draught, tons	14,158
Coal carried on trials, tons	1,242
Full capacity of bunkers, tons	2,300
Radius of action at 10 knots, miles (designed)	12,000

The hull is of mild steel throughout, and is provided with docking and bilge keels. A belt of side armor extends from end to end of the vessel, and is connected with two armored decks. The belt armor is 12 feet 2 inches deep amidships, where it is 7 inches thick, this thickness being reduced to 4 inches at the two ends. Two athwartship armored bulkheads 7 inches thick extend from the side plating to the center barbettes. The splinter deck is flat, and varies from 24 mm. to 34 mm. (.94 inch to 1.33 inch) thick. The armored deck is partly sloping; where flat it is 45 mm. (1.77 inches) thick and where sloping 65 mm. (2.56 inches) thick. All round the ship a cofferdam has been worked behind the side belt armor.

The armament consists of fourteen 7.5-inch guns, as follows: Two in the forward and astern central turrets; six in turrets arranged three on each side of the vessel; and four in casemates, two forward and two aft. There are, in addition, eighteen 65 mm. (2.56 inches) guns and eight 47 mm. (1.85 inches) guns. Nine of the 7.5-inch guns can be fired on the broadside and four ahead and astern. There are two 18-inch torpedo tubes.

The thickness of the armor in the turrets is 6.5 inches, and their splinter plates are 4 inches thick. The armor of the bar-

bettes is also 6.5 inches thick, their floor plates 1.6 inches, and their inner tube 3.2 inches. The outside plating of the casemates is 6.5 inches, the inside plating 4 inches, and their splinter plates 3.5 inches. The conning tower, which is on the lower deck, is made of 10-inch plating, and has an 8-inch shield.

The vessel is steam-heated throughout, and nearly all her auxiliaries are electrically driven. There are forty Delaunay-Belleville water-tube boilers working at 300 pounds pressure, and having a total grate area of 231 square meters—about 2,486.5 square feet—and a heating surface of 7,605 square meters—about 81,860 square feet.

The propelling machinery consists of three sets of triple-expansion four-cylinder engines, designed to develop 36,000 indicated horsepower at a speed of 136 revolutions per minute, and this power was to produce a speed of 23 knots.

Data of the official trials are set forth below:

	Actual results.	Contract figures.
Ten Hours' Full-Power Trial:		
Number of boilers	40	...
Mean indicated horsepower	36,823	36,000
Coal consumption per I.H.P. hour, pounds	1.675	1.82
square foot of grate area, lbs.	25.38	...
Mean speed for the 10 hours, knots	23.8	23
Three Hours' Full-Power Trials with Three-quarters of Boilers:		
Number of boilers	30	...
Mean indicated horsepower	39,803	36,000
Coal consumption per I.H.P. hour, pounds	1.931	...
square foot of grate area, lbs.	39.31	...
Mean speed for the 3 hours, knots	23.92	23
Twenty-four Hours' Consumption Trial:		
Number of boilers	40	...
Mean indicated horsepower	25,416	24,000
Coal consumption per I.H.P. hour, pounds	1.344	...
square foot of grate area, lbs.	14.03	...
Mean speed, knots	21.96	...
Six Hours' Consumption Trial:		
Number of boilers	20	...
Mean indicated horsepower	16,705	16,000
Coal consumption per I.H.P. hour, pounds	1.29	1.543
square foot of grate area, lbs.	16.71	...
Mean speed, knots	19.55	...

Condorcet.—The speed trials of the battleship *Condorcet* were carried out on the 13th January. This vessel has a displacement of 18,000 tons, and a shaft horsepower of 22,500. She is fitted with twenty-six Niclausse boilers of the latest improved type, made by Messrs. J. & A. Niclausse of Paris, and eight turbines built in a French dockyard. This vessel is the first French *Dreadnought* to carry out speed trials. The full-power trial, as provided for in the contract, consisted of a run of 10 hours with all boilers in use, and burning 26½ pounds of coal per square foot of grate surface per hour. The result of the trial gave an average speed under the above conditions of 19.257 knots as against 19.25 knots provided for. The Naval Ministry further asked Messrs. J. & A. Niclausse if they would consent to a full-power trial with a combustion of 33.34 pounds of coal per square foot of grate, which was accepted. The trial under these conditions was carried out on the 16th January, and the average speed attained over a 10-hours run was 19.80 knots, more than half a knot over the contract speed. The *Condorcet* further carried out her endurance trials on the 20th and 21st January. This trial consisted of a run of twenty-four hours with normal combustion, and the average speed attained was 18.2 knots with a coal consumption per mile lower than was allowed for in the contract. The performance of the turbines and boilers was perfectly satisfactory throughout all the trials; the desired steam pressure being easily maintained under all conditions.—" The Steamship."

GREAT BRITAIN.

Battleship.—Neptune.—This vessel, the hull of which was built at Portsmouth dockyard, has successfully completed her trials. She is an improved *Dreadnought* type, propelled by Parsons turbines, these latter having been constructed by John Brown & Co., Clydebank.

As regards the gun trials, the feature of interest was the

effect of the firing of guns with their muzzles almost exactly over the turrets of the guns in front; for the two turrets aft were thus arranged, the guns of the one to the rear and at a higher level than those of the turret in front. No inconvenience was caused by the blast, and thus the four guns in these two turrets, as well as the two guns on each broadside, may be fired simultaneously astern. The gun mountings, of the hydraulic type, for ensuring accuracy and rapidity of fire gave full satisfaction.

As regards the speed performance, it is interesting that the *Neptune* is the first battleship tried without cruising turbines. In place of these separate units fitted in many recent ships, the ordinary high-pressure turbine is lengthened. At low power the steam does useful work for the full length of the high and low-pressure turbines, by which a satisfactory range of expansion is got; but at higher powers the steam is admitted at an intermediate stage in the length of the turbine with economical results. Progressive trials, each of four hours, were run at powers ranging from 1,800 up to 26,000 shaft horsepower. During part of each trial data were taken of the economy—(1) with the auxiliary machinery exhausting into the auxiliary condenser, and (2) with such exhaust steam passing into the low-pressure turbine. The great advantage of the Parsons low-pressure turbine in the latter case was clearly demonstrated. It was shown that, using only such exhaust steam from auxiliaries, the *Neptune* could travel at least five knots. When the auxiliary exhaust was passed to the auxiliary condenser the power on one trial was 3,662 shaft horsepower, but when the exhaust went into the turbine the power increased to 5,069 shaft horsepower, without any additional demand upon the boilers. There was a corresponding increase on another trial from 6,911 to 8,579 shaft horsepower, and on others from 13,438 to 16,513, and from 23,579 to 26,836 shaft horsepower. The fuel economy was correspondingly increased.

On the thirty hours' trial at 70 per cent. of the full power the turbines developed 18,373 shaft horsepower, and the speed on the measured mile was 19 knots; while on the full power eight-

hours' trial, the mean power was 27,721 shaft horsepower, and the speed on the mile 21.786 knots.

It is worthy of note that this ship has been completed for sea and actually commissioned a week under two years from the date that her first keel plate was laid.

H. M. Torpedo-Boat Destroyer Hope.—The trials of the this vessel—one of the latest of the destroyers built for the British Admiralty—have quite recently been carried through successfully.

The *Hope* belongs to what is known as the *Acorn* class, of which twenty were ordered under the Navy Program of 1909–10. These vessels are from the designs of Sir Philip Watts, K.C.B., the Director of Naval Construction, and vary little in details of construction. Their length is 240 feet, their beam 25 feet 3 inches, and their draught 7 feet 9 inches, when the displacement is about 780 tons. They are fitted with two tubes for firing torpedoes, and their armament includes two 4-inch quick-firing guns and two 12-pounders. They are thus more powerfully armed than their predecessors, and are, moreover, much more strongly built, with high forecastles in order to maintain full speed in heavy seas.

The propelling machinery is of the Parsons reaction turbine type, and in all there are seven turbines; a high-pressure cruising, an intermediate cruising, a high-pressure main turbine, and two low-pressure turbines, these turbines all being for propelling the vessel ahead; in addition, there are two turbines for astern-going purposes, and these are incorporated in the casings of the ahead-going low-pressure turbines. There are three lines of shafting, and mounted on each wing shaft are one cruising, one low-pressure ahead and one astern turbine. The high-pressure turbine only is connected to the center line of shafting.

For cruising purposes, during which periods a low power is required, the steam is passed in series through the whole of the ahead-going turbines, commencing with the high-pressure cruising turbine, and it is under these conditions that the economical advantages of these cruising turbines becomes ap-

parent. For full speed both cruising turbines become inoperative units so far as propulsion is concerned. The low-pressure ahead and astern turbine alone are used in maneuvering, the center turbine being idle.

For generating the steam there are four express water-tube boilers of the Yarrow type. They are the design and arrangement now usually adopted for this class of vessel. The fuel used for the boilers is a heavy oil, and that used for the trials was of the usual quality as supplied to the Admiralty.

The principal trial consisted of an eight-hours' run at full power, with six runs over the measured mile; these latter runs were made on the St. Abb's course. No attempt was made to exceed largely the designed speed of 27 knots, and the mean power developed was about 15,000 shaft horsepower, the mean revolutions of the turbines being 737 per minute. On the eight-hours' trial the results were equally satisfactory. An important feature was the easy maintenance of steam, and the fact that the vacuum averaged 28½ inches throughout. A high fuel economy was attained, and the results throughout were most favorable.

H.M.S. Larne, the first of the four destroyers built by Messrs. John I. Thornycroft & Co., Limited, at their Woolston Works, of the 1909 program, was commissioned recently at Portsmouth after completing the necessary handing-over trial.

The H.M.S. *Larne* is the first of the English-built destroyers of this program to be delivered, and Messrs. Thornycroft launched on the 2d of February the last of the four vessels, viz: H.M.S. *Minstrel,* which vessel was launched complete with machinery and boilers on board.

The second vessel of the type, H.M.S. *Lyra,* will soon be finished, and it is expected that the third vessel, H.M.S. *Martin,* will be completed ready for commissioning before the end of March, while the fourth vessel, H.M.S. *Minstrel,* will be completed ready for commissioning before the end of April.

These destroyers have a displacement of 780 tons, and are propelled with Parsons turbines of sufficient power to give a speed of 27 knots. The boilers are fitted to work with oil fuel.

HOLLAND.

Wolf.—A successful official full-speed trial of the *Wolf*, one of the two destroyers ordered by the Dutch Government from Messrs. Yarrow & Co., of Glasgow, and constructed by the Koninklijke Maatschappij "De Schelde," of Flushing, embodying Messrs. Yarrow's latest and most improved practice, took place on the 26th January at the mouth of the Schelde with highly satisfactory results. A mean speed of 30.08 knots was easily obtained during the three-hours' trial, this being well in excess of the contract speed.

The other destroyer, the *Fret*, has also passed through her official speed trials with satisfactory results.

MERCHANT SHIPS.

Olympic.—The White Star liner *Olympic,* the largest vessel thus far constructed, was launched from Messrs. Harland & Wolf's yard at Belfast on October 20, 1910. This vessel and her sister ship, the *Titanic,* have the following principal dimensions:

Length over all, feet and inches	882–09
between perpendiculars, feet	850
Breadth, extreme, feet	92
Depth, molded, feet and inches	64–03
Total height from keel to navigating bridge, feet	104
Gross tonnage, tons	45,000
Load draught, feet and inches	34–06
Displacement at load draught, tons	about 60,000
Indicated horsepower of reciprocating engines	30,000
Shaft horsepower of turbine engine	16,000
Speed, knots	21

The propelling machinery is composed of a combination of reciprocating engines and turbines arranged on three shafts. Each wing shaft is driven by a 4-cylinder, triple-expansion, vertical, inverted, reciprocating engine with dimensions as given below.

Diameter, H.P. clyinder, inches	54
I.P. cylinder, inches	84
L.P. cylinder, inches	97
Stroke, inches	75
Diameter, crank shaft, inches	27
Boiler-working pressure, pounds per square inch, gage	215

The engines have link-valve motion, and piston valves to the high and intermediate cylinders, and slide valves to the low-pressure cylinders. The engines are 39 feet from base plate to the highest point.

There are twenty-nine boilers, of which twenty-four are double-ended, 15 feet 9 inches diameter by 20 feet long, and five single-ended, 15 feet 9 inches diameter by 11 feet 9 inches long. The boilers will be placed in six boiler rooms separated by bulkheads. In the first boiler room there will be the five single-ended boilers; in the second, third, fourth and fifth, five double-ended boilers, and in the sixth boiler room four double-ended boilers.

The center screw is driven by a Parsons exhaust turbine designed to work ordinarily with an initial steam pressure (exhaust from reciprocating engines) of 9-pounds absolute per square inch and a vacuum of 28½ inches of mercury. In maneuvering the wing shafts only are used, the exhaust steam from the reciprocating engines on such occasions being sent directly, by means of change valves, to the condensers.

OBITUARY.

DAVID B. MACOMB.

Chief Engineer David B. Macomb, U. S. Navy, son of General Alexander Macomb, one of the first Adjutants General of the Army, was born in Florida in 1827, and entered the Navy as a Third Assistant Engineer in 1849. He died at his home in New York City on the 27th of January, 1911, being 84 years of age, and having been 62 years in the Navy. He is survived by his wife, a daughter of Commodore John Pope, and by two daughters, who married officers of the Navy.

Mr. Macomb's early training was with the firm of Merrick and Towne, of Philadelphia, where he acquired proficiency in both the theoretical and the practical side of his profession. His ability was recognized in his early assignment to duty in the office of the Engineer-in-Chief, where he was engaged on the design of machinery for the *Susquehanna,* the *San Jacinto* and the *Saginaw,* a service which materially retarded his promotion, for at that time the regulation requiring a certain sea service before promotion was rigorously enforced.

Subsequently, he served on board the Coast Survey Steamer *Bibb,* and on the exploring ship *John Hancock,* the latter in the North Pacific, through the Aleutian archipelago and on the coasts of China and Japan. He afterwards served on board the frigate *Wabash* and the *Saranac,* and reached the grade of Chief Engineer in 1860, when he was ordered to the frigate *Niagara,* taking the Japanese Embassy back to their native country. During the Civil War Mr. Macomb was in the attack on the Pensacola Navy Yard, at the reduction of Fort McRae and other places in the Gulf. He was an early advocate of iron ships and iron clads, particularly the monitors, and in the early days of those vessels was considered an authority

on their construction and management. He superintended the construction of a number of them, and was serving on board the monitor *Canonicus* when she reduced the famous Howletts Battery, and when she had the encounters with Dutch Gap, Deep-Bottom, Fort Fisher and the batteries of Charleston. The *Canonicus* went in chase of the famous Confederate ram *Stonewall,* and followed her into Havana, being the first U. S. iron clad to enter a foreign port.

After that war Mr. Macomb served in the Navy Yards at Pensacola and Portsmouth, N. H.; then Fleet Engineer of the North Atlantic Squadron, and at the Boston Navy Yard; and on the statutory Board of Appraisal and Survey.

He was a member of the American Society of Naval Engineers, of the American Society of Mechanical Engineers, of the Loyal Legion, of the Union League Club of New York, and of the Freemasons, and had a host of warm friends in all of them. He was conspicuous by his charm of manner and his devotion to his friends, the distinguishing trait in his character being his consideration for others. He was one of the last of the "Old Chiefs" who have stood the heat and burden of the day, and in his passing the Navy loses one of its landmarks.

G. W. B.

JARVIS BONESTEEL EDSON.

Mr. Edson was born in Wisconsin and died at his home in New York City on the 26th of January, 1911. He was educated at New York University and served in the Navy as a Third Assistant Engineer from November, 1864, to August, 1868, when he resigned.

His service in the Navy during the Civil War was on board the *Fah-Kee* and the *Cambridge,* which vessels were on the blockade off the coasts of North Carolina and of South Carolina. When the war was over he served on board the *Wasp,* in the South Atlantic Squadron, along the coasts of Paraguay, Uruguay and Brazil.

He contributed a number of useful inventions to engineering, among which were the recording steam gage; the process of drying gun-cotton in compressed air, at ordinary temperatures; the method, now so generally used, of sinking deep wells in quicksand and in clay. He was for some years engaged in the manufacture of celluloid and zylonite, in Newark, N. J., and in North Adams, Mass.

He was a member of the Engineers' Club, in New York; of the American Society of Naval Engineers; the New York Commandery of the Naval Order; the American Society of Mechanical Engineers; the Society of Naval Architects and Marine Engineers; the New York Yacht Club; the Shelter Island Yacht Club; the Columbia Yacht Club, and the Sons of the Revolution.

He was a very good after-dinner speaker, and a quick and ready debater, and though much interested in local politics was never actively so; he saw the good, rather than the bad, in all candidates, and was sanguine as to prospective results, therefore, not the kind of man to be active in politics.

He married Miss Eliza Robins, of Brooklyn, who, with two sons, survive him. Since his retirement from business a few years ago, he has spent his winters in his New York home, but his summers were divided between his Shelter Island cottage and cruising in his steam yacht, the *Viking*.

He was a lover of music and of the dramatic art, and of outdoor sports. He spent a great deal of money in charity, but never in an ostentatious way. He was much loved by all who knew him and his demise will be felt for a long time.—G. W. B.

BOOKS RECEIVED.

THE ELEMENTS OF GRAPHIC STATICS AND OF GENERAL GRAPHIC METHODS. By WILLIAM LEDYARD CATHCART and J. IRVIN CHAFFEE. Published by the D. VAN NOSTRAND COMPANY, No. 23 Murray and No. 27 Warren Streets, New York. Price $3.00, net.

This book has been prepared for the use of students of mechanical and marine engineering. Graphic methods have had their widest application in the analysis of the stresses in stationary structures, and in consequence, most text-books on the subject have been written for civil engineers. In this work, however, graphic statics is handled from the mechanical engineer's viewpoint, and for that reason it is one of the relatively few books in the English language covering the subject in this way.

The contents of the eleven chapters cover the whole subject of graphical methods; hence, the scope of the book is wide, while the number of pages is but 300. The treatment is, therefore, very concise.

The fundamental principles of roof and bridge trusses are discussed at some length; this is usually considered as structural and not mechanical engineering. A walking beam is, however, a braced cantilever, which is designed like a roof truss; and a mechanical engineer, in designing the girder for an overhead crane, must know the principles governing live loads and impact, as in a bridge truss.

The work is clearly written, profusely illustrated throughout, and the contents cannot fail to prove of great interest and value to those who have to deal with engineering problems.

ASSOCIATION NOTES.

SEVENTIETH BIRTHDAY OF ADMIRAL MELVILLE.

To a man of heroic mould, who has occupied a foremost position in governmental and professional circles, it might almost seem, as the years go by, that nothing remains which can give a special charm to any occasion. There is, however, one thing which most of us as we grow older come to prize above all else, namely, the friendship and affection of our fellow workers, and when this is manifested in a striking way it probably gives a pleasure to the recipient which cannot be equaled in any other manner.

That grand old man of engineering, Admiral Melville, who for the longest period in the history of the Navy Department was Chief of a Bureau, and who was the exponent of engineering during the early days of the building of the new Navy, reached his seventieth birthday on January 10th of this year. It had occurred to some of his friends who are in close touch with him and who knew the date, to remind others of its approach, so that there could be a concerted plan of sending him letters of affectionate greeting and good wishes when the day came. These reminders went out about Christmas of last year, but at once many of the recipients suggested that something more striking be done, and it was at first proposed to have a dinner in New York. The Admiral did not feel that he was quite equal to this in view of his bereavement not so many months ago, and declined with sincere thanks. It then occurred to Colonel E. D. Meier, President of the American Society of Mechanical Engineers, that by quick action a suitable piece of silver plate might be secured in time for presentation on the birthday. A small committee was formed, consisting

1841 1911

Rear Admiral
George Wallace Melville,
Engineer in Chief U. S. Navy
(Retired)

After a long voyage, replete with adventures and hardships, adverse tempests and uncharted currents, you are now passing the

Seventieth Meridian of Your Life's Journey.

We, your friends, unite in wishing you a mellow evening, smooth seas and favoring breezes. Looking back over the bright wake you have traced on the ocean of life, we see you

First, as a stripling, a leader in torpedo attack, oil bottle on, in stoke hole, pushing the ship into action at the close range then necessary.

Next, in your young manhood thrice braving the frozen North in works of rescue;

On your prime, designing and perfecting [redacted] propelled the ships that aided Manila and Santiago to the laurels of our navy.

Even now inventing the mechanism which makes possible the universal combination of turbine and screw.

Your Success

the natural result of keen insight, clear reasoning, devotion to duty, undaunted truth, and courage which carried pathways through arctic ice and made vibrant with life the very metal of your engines.

Your warm appreciation of the work for science and co-operation, your sinking the man in the work, have endeared you to all who know you, and are an inspiration to every young engineer.

Allow us to present you with a modest but enduring token of our admiration, and love for you, on which the engravers art has indicated a few of the achievements which have made you the model and type of the

American Engineer.

[redacted] JANUARY TENTH, NINETEEN ELEVEN

[illegible] Meier	John Fritz	George Westinghouse Jr.	J. Coes	George W. Quintard	W. A. Wiley
[illegible] Upjohn	Stevenson Taylor	Chas. T. Ely	C. J. Manning	[illegible] Leavitt	[illegible] Wallace Hunt
Alex. C. Humphreys	A. M. Mattoes	[illegible] E. Palmer	[illegible] Wells	[illegible] A. Wells	[illegible] Worth
[illegible] Deane	John Thomson	[illegible] M. Perry	[illegible] Verity	[illegible] Worthington	[illegible] Mc Kinney
[illegible] Taylor	Joseph Morgan	Jesse M. Smith	[illegible] Griffin	[illegible] Wadhe	[illegible] Wade
[illegible] Allen	George W. Magee	[illegible] Bailey	[illegible] Bryan	[illegible] Hutton	[illegible] Corbitt
Theo. Smith	[illegible] Clark	Warner Mills	[illegible] Carr	[illegible] Avery	[illegible] Palmer
[illegible] Alverson	[illegible] Jacobus	[illegible] Mc Vay	[illegible] Dyson	Charles A. Moore	Jos. M. Read
[illegible] Davidson	[illegible] Reid	[illegible] Marshall	[illegible] Durack	[illegible] Andreaus	[illegible] Hawkins
[illegible] Dyson	[illegible] Wagoun	[illegible] Miller	[illegible] Mc Farland	Joseph N. Church	[illegible] Dingelbury
[illegible] Greene	[illegible] Sheere	[illegible] Moore	[illegible] Norton	[illegible] Meyer	[illegible] Gadsby
Wm. Mc Conway	[illegible] Rice	[illegible] Osborne	[illegible] Reed	Edwin Mills	Robinson
Albert Morris	[illegible] Worden	[illegible] Hyatt	Emil Heine	[illegible] Raynal	Pratt
John Platt	[illegible] Bunker	Moses Webster	[illegible] Cooley	John [illegible] Stanton	[illegible] Stratton
[illegible] Smith	George B. Read	[illegible] W. Hunt	Chas. A. Perry	Charles Ward	[illegible] Wurts
[illegible]	[illegible] Fraser	[illegible] Ward		[illegible] Williamson	[illegible] Ward

With Greetings to
Rear Admiral George Wallace Melville
Engineer in Chief
United States Navy (retired)
From His Many Friends on the
Seventieth Birthday
January Twenty
191

of Colonel Meier, Mr. E. G. Spilsbury, Colonel R. C. McKinney, Mr. H. L. Aldrich and Mr. W. M. McFarland, who quickly advised about 100 of the Admiral's personal friends of what it was proposed to do, and, feeling sure that there would be a unanimous response, arranged with Dr. George F. Kunz, of Tiffany's, for the plate, a photographic reproduction of which is given. The replies to the letter of invitation were as flattering as any man could wish to read. Nearly all of them expressed thanks to the committee for the opportunity to join in this tribute of respect.

Meanwhile it was considered desirable that there should be something in the form of an address to accompany the piece of plate, and accordingly the beautiful tribute which is shown on the opposite page was prepared by Colonel Meier and most heartily approved by all to whom it was submitted.

The work of preparing both plaque and address had to be done in about a week, but all who had any part in it seemed to share the enthusiasm of the donors, with the result that on the morning of the 10th of January they were ready. At the suggestion of the Committee and other donors, the plaque and address were taken over to Philadelphia by Mr. Asa M. Mattice and Mr. Walter M. McFarland, both formerly assistants to Admiral Melville while Engineer-in-Chief, and were presented to him in person by them. We have learned from these gentlemen that the Admiral was deeply touched at the evidence of the affectionate regard in which he is held by so many of his friends, and found it almost impossible to formulate his thanks.

Besides these beautiful evidences of the esteem in which he is held by his friends, the Council of the American Society of Mechanical Engineers sent him a telegram of congratulations, and he was also advised of his election as an honorary member of the Engineers Club of New York, which is a signal mark of respect, as there have only been five honorary members thus far in the history of the club, the late Mr. Haswell, Mr. Carnegie, Mr. John Fritz, Admiral Melville and Mr. Thomas A. Edison.

It is peculiarly appropriate that this account of the way in which Admiral Melville's birthday was honored should appear in this JOURNAL, because it was largely owing to his interest and fostering care that the Society of Naval Engineers made its start and the JOURNAL attained its high standing in engineering literature.

Undoubtedly a great many others would have been delighted to join in this testimonial if the opportunity had been afforded them. We have learned, however, from the Committee that the time was so short that they were compelled to limit the invitations to those who were readily accessible.

It will be seen in the list of signatures to the address that about 20 per cent, are naval officers; it is also extremely interesting to note that so many of the leaders of the profession are included. Such a striking testimonial shows the high regard felt for Admiral Melville by the civilian as well as the naval members of the profession.

THE ANNUAL BANQUET OF THE SOCIETY, which was a pronounced success, was held at Rauscher's, Washington, D. C., on February 11, 1911. In addition to entertainment in a lighter vein, toasts were responded to, as given below:

Toasts.

Toastmaster, Mr. WALTER M. MCFARLAND.

"The Navy," HON. ERNEST W. ROBERTS.

"The Future of the Navy," HON. L. P. PADGETT.

"Engineering in Connection with Fleet Operations," REAR ADMIRAL R. WAINWRIGHT, U. S. N.

"The Navy and the People," COL. ROBT. M. THOMPSON.

Members and Guests Attending.

Paymaster David M. Addison, U.S.N.
Mr. H. L. Aldrich.
Comdr. Philip Andrews, U. S. N.
Mr. J. F. J. Archibald.
Lt. Comdr. Marques de Azevedo, Brazilian Navy.
Lt. Comdr. Benoit d'Azy, French Navy.
Mr. R. O. Bailey.
Rear Admiral Baird, U. S. N.
Lt. Comdr. F. P. Baldwin, U. S. N.
Dr. Edmund Barry.

ANNUAL BANQUET OF THE SOCIETY OF NAVAL ENGINEERS AT RAUSCHER'S, WASHINGTON, D. C., FEBRUARY 11, 1911.

Capt. F. W. Bartlett, U. S. N.
Capt. F. E. Beatty, U. S. N.
Mr. Ira E. Bennett.
Mr. C. K. Berryman.
Mr. C. C. Billings.
Mr. Scott Bone.
M. T. W. Braheny.
Mr. Joseph C. Breckons.
Lt. H. E. Brinser, U. S. N.
Mr. George K. Bradfield.
Capt. B. C. Bryan, U. S. N.
Dr. Edgar Buckingham.
Mr. William Callahan.
Asst. Engineer R. E. Carney, U.S.N.
Comdr. L. H. Chandler, U. S. N.
Mr. R. H. Chappell.
Lt. H. T. Church, U. S. N.
Rear Admiral C. E. Clark, U. S. N.
Mr. Winthrop Cole.
Paymaster General Cowie, U. S. N.
Lt. O. L. Cox, U. S. N.
Mr. Chas. Francis Coffin.
Mr. R. H. Danforth.
Mr. H. E. Davis.
Mr. Oscar King Davis.
Hon. Albert F. Dawson.
Commodore R. G. Denig, U. S. N.
Mr. George H. Diman.
Lt. W. G. Diman, U. S. N.
Mr. W. E. Dowd, Jr.
Comdr. C. W. Dyson, U. S. N.
Lt. P. B. Eaton, U. S. R. C. S.
Hon. W. T. Englebright, M. C.
Capt. A. T. Fechteler.
Mr. F. C. Fergusson.
Lt. J. O. Fisher, U. S. N.
Hon. John J. Fitzgerald.
Capt. F. F. Fletcher, U. S. N.
Mr. C. B. Ford.
Mr. W. B. Gifford.
Mr. C. P. Gliem.
Hon. Alex. W. Gregg, M. C.
Mr. H. B. Gregory.
Capt. R. S. Griffin, U. S. N.
Lt. Norman Hall, U. S. R. C. S.
Lt. Comdr. John Halligan, U. S. N.
Mr. R. D. Heinl.
Mr. James E. Henry.
Mr. Fred. G. Herbert.
Mr. H. L. Hibbard.
Hon. Charles D. Hilles.
Comdr. Tokotura Hiraga, Japanese Navy.
Civil Engineer Holliday, U. S. N.
Prof. Ira N. Hollis.
Lt. Comdr. U. T. Holmes, U. S. N.
Mr. William Hoster.
Ensign J. B. Howell, U. S. N.
Comdr. C. F. Hughes, U. S. N.
Mr. William J. Hughes.
Mr. Lewis Jones.
Mr. Ernest N. Jansen.
Mr. J. E. Jenks.
Mr. Arthur C. Johnston.
Comdr. L. A. Kaiser, U. S. N.
Mr. L. Kalzenstein.
Mr. Martin L. Kalzenstein.
Mr. William Kalzenstein.
Mr. C. G. T. King.
P. A. Eng. William R. King, U.S.N.
Comdr. T. W. Kinkaid, U. S. N.
Lt. Scudder Klyce, U. S. N.
Capt. J. J. Knapp, U. S. N.
Lt. H. Kotzschmar, U. S. R. C. S.
Mr. Marshall Q. Leighton.
Mr. S. Inglis Leslie.
Mr. J. S. Leslie.
Mr. Luther D. Lovekin.
Hon. George Alvin Loud, M. C.
Ch. Engineer G.W. Magee, U. S. N.
Lt. S. I. M. Major, U. S. N.
Ch. Eng. Chas. H. Manning, U.S.N.
Mr. A. M. P. Maschmeyer.
Mr. H. A. Magoun.
Mr. A. M. Mattice.
Mr. Jerry Matthews.
Comdr. W. J. Maxwell, U. S. N.
Engineer-in-Chief C. A. McAllister, U. S. R. C. S.
Dr. Bernard McCullom.
Mr. Walter M. McFarland.
Col. E. B. Meier.
Adm'l George W. Melville, U. S. N.
Paymaster J. H. Merriman, U. S. N.
Mr. A. B. LeP. Mesny.
Mr. J. F. Metten.

Mr. W. R. Metz.
Mr. James Mewshaw.
Mr. Charles Mulliken.
Mr. B. C. Moise.
Lt. Comdr. R. C. Moody, U. S. N.
Mr. H. T. Morningstar.
Mr. Richard Morton.
Mr. J. H. Mull.
Judge Mullowney.
Mr. William Murray.
R. Adm'l R. O. Nicholson, U. S. N.
Mr. J. H. Nicholson.
Ensign W. C. Nixon, U. S. N.
Lt. Albert Norris, U. S. N.
Lt. Wm. Norris, U. S. N.
Mr. L. B. Nutting.
Mr. George A. O'Connor.
Mr. J. Callan O'Loughlin.
Hon. J. Van Vechten Olcott, M. C.
Hon. Lemuel P. Padgett, M. C.
Mr. E. B. Paine.
Capt. W. M. Parks, U. S. N.
Comdr. J. B. Patton, U. S. N.
Senator Thos. H. Paynter.
Capt. H. S. Pettus, U. S. A.
Lt. Comdr. Carlo Pfister, Italian Navy.
Rear Admiral Potter, U. S. N.
Captain Templin C. Potts, U. S. N.
Mr. Henry S. Powell.
Mr. J. W. Powell.
Comdr. Baron Preuschen, Austrian Navy.
Mr. Pitman Pulsifer.
Mr. Joseph E. Ralph.
Lt. Comdr. M. E. Reed, U. S. N.
Commodore I. S. K. Reeves, U.S.N.
Comdr. Retzman, German Navy.
Lt. J. O. Richardson, U. S. N.
Hon. Ernest W. Roberts, M. C.
Lt. S. M. Robinson, U. S. N.
Lt. Comdr. J. K. Robison, U. S. N.
Comdr. S. S. Robison, U. S. N.
Mr. R. R. Row.
Mr. H. W. Rowley.
Mr. P. J. Ryan.
Lt. Comdr. W. R. Sexton, U. S. N.
Mr. George C. Shepard.
Mr. W. F. Sicard.
Hon. C. Bascom Slemp, M. C.
Mr. Hal H. Smith.
Capt. Roy C. Smith, U. S. N.
Comdr. W. Strother Smith, U. S. N.
Capt. Thomas Snowden, U. S. N.
Major Stanley, U. S. A.
Surgeon-General Stokes, U. S. N.
Comdr. S. W. Stratton, D. C. N. M.
Rear Admiral Wm. Swift, U. S. N.
Lt. Comdr. W. B. Tardy, U. S. N.
Mr. Elisha Theall.
Col. Robert M. Thompson.
Hon. R. J. Tracewell, Compt. Treas.
Comdr. Emil Theiss, U. S. N.
Gen. Geo. Uhler. Steam. Insp. Serv.
Comdr. Vassilieff, Russian Navy.
Mr. Leroy Y. Vernon.
Mr. Mercer Vernon.
Rear Admiral Wainwright, U. S. N.
Mr. Ernest G. Walker.
Mr. George E. Walton.
Constr. J. Q. Walton, U. S. R. C. S.
Mr. Charles Ward.
Chief Constructor Watt, U. S. N.
Hon. John W. Weeks, M. C.
Lt. Comdr. W. B. Wells, U. S. N.
Mr. George Westinghouse.
Comdr. W. W. White, U. S. N.
Mr. Elliott Woods.
Lt. N. H. Wright, U. S. N.
Lt. Comdr. D. W. Wurtzbaugh, U.S.N.
Lt. T. H. Yeager, U. S. R. C. S.

AT THE ANNUAL MEETING OF THE SOCIETY held at the Navy Department, Washington, D. C., December 30, 1910, the following were elected officers for the ensuing year:

President, Engineer-in-Chief H. I. Cone, U. S. Navy.

Secretary-Treasurer, Lieutenant Commander U. T. Holmes, U. S. Navy.

Council, Captain R. S. Griffin, U. S. Navy; Engineer-in-Chief Charles A. McAllister, U. S. R. C. S.; Lieutenant Commander J. K. Robison.

A statement showing receipts and expenditures for the year 1910 is given below:

ANNUAL STATEMENT, JANUARY 1, 1911.

Receipts.		
Dues and subscriptions	$5,566.03	
Advertisements	1,085.23	
Sales	504.93	
Interest on bonds and bank balance	313.55	
Subscriptions to annual banquet	530.00	
		$7,999.74
Expended.		
Printing and stationery, including postage on regular issues	$4,895.61	
Engraving	632.28	
Drafting	48.50	
Salary, Secretary-Treasurer	900.00	
Employment of clerk	250.00	
Manuscript paid for	904.00	
Cancelled subscriptions	40.00	
Prize essay	50.00	
Gold medal to prize essayist	30.00	
Commission on advertisements	33.76	
sales	9.37	
Express and postage	93.27	
Stationery	45.55	
Miscellaneous expenses	21.75	
Safe deposit box	5.00	
Expenses of annual banquet	883.70	
		$8,842.79
Available assets.		
Cash balance in bank, January 1, 1911	$2,856.23	
Six Washington Railway and Electric 4 per cent. bonds at 86	5,160.00	
Total		$8,016.23

Cash balance, January 1, 1910	$3,699.28	
1911	2,856.23	
Loss during year		$843.05

The above statement was audited by a committee of the Council and found correct.

W. W. WHITE,
Commander, U. S. Navy,
Acting Secretary-Treasurer.

The Society's membership (January 1, 1911.) is as follows:

Honorary members,	6
Prize essayists,	9
Members,	358
Associates,	283
Subscribers,	324
Exchanges,	69

Attention is invited to the desirability of increasing the membership. The Council will appreciate all efforts on the part of Members and Associates in bringing to the attention of those eligible the important engineering data published in the Society's JOURNAL.